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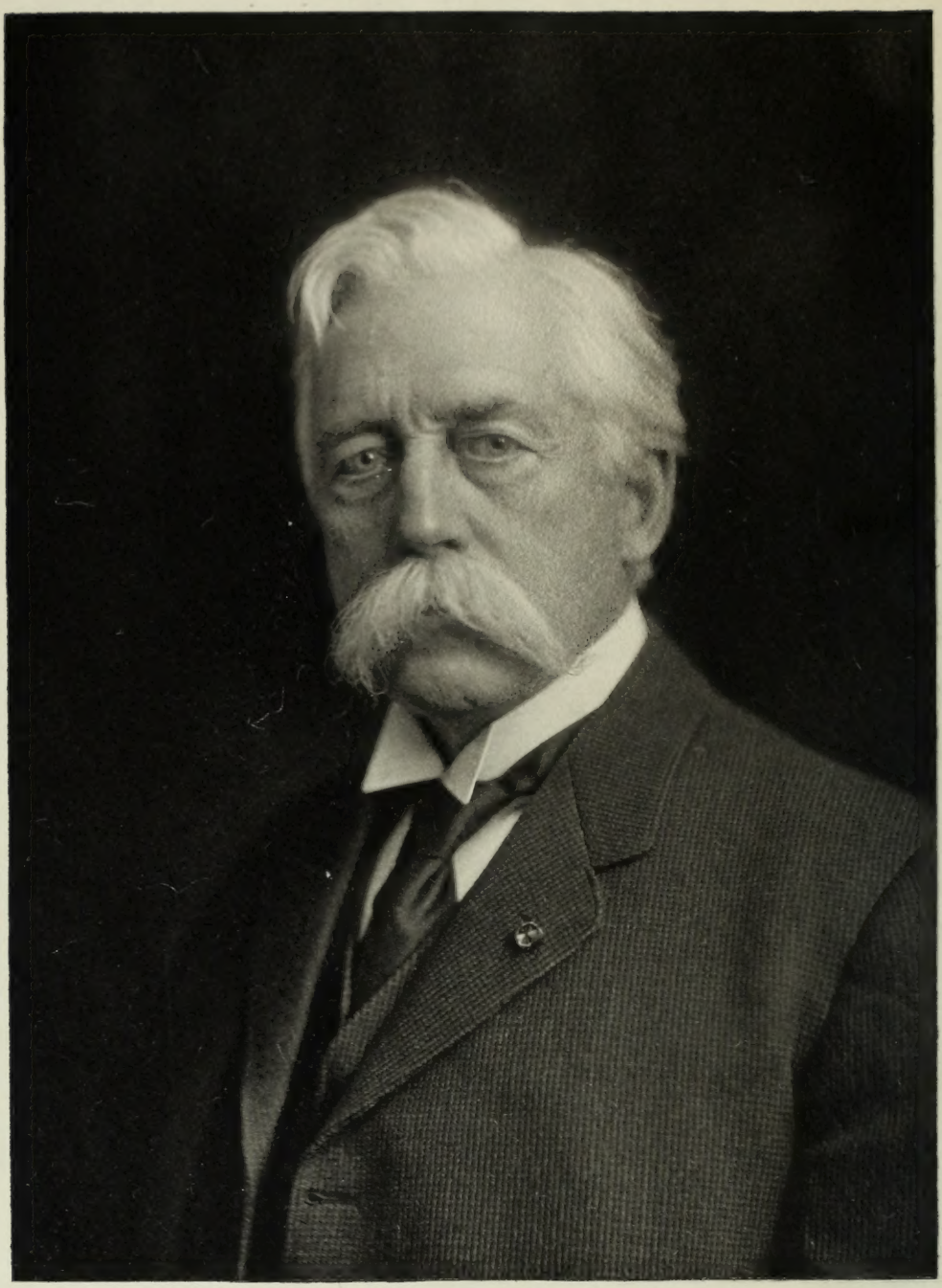

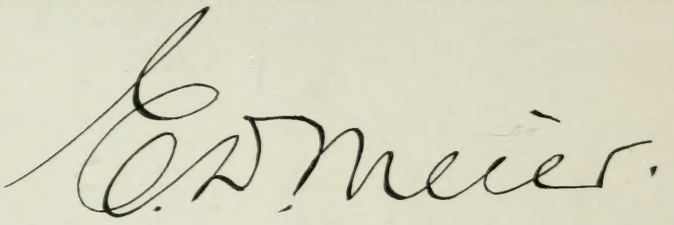

PRESIDENT 1911
OF
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 33

PITTSBURGH MEETING
NEW YORK MEETING
1911

124796
7/11/12

NEW YORK
PUBLISHED BY THE SOCIETY
29 West 39th Street
1912

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1911

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E. D. Leavitt	1883	Cambridge, Mass.
John E. Sweet	1884	Syracuse, N. Y.
J. F. Holloway	1885	Died September 1, 1896
Coleman Sellers	1886	Died December 28, 1907
George H. Babcock	1887	Died December 16, 1893
Horace See	1888	Died December 14, 1909
Henry R. Towne	1889	New York, N. Y.
Oberlin Smith	1890	Bridgeton, N. J.
Robert W. Hunt	1891	Chicago, Ill.
Charles H. Loring	1892	Died February 5, 1907
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S. T. Wellman	1901	Cleveland, O.
Edwin Reynolds	1902	Died February 19, 1909
James M. Dodge	1903	Philadelphia, Pa.
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1911

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F. R. Low (4)
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Note—Numbers in parentheses indicate number of years the member has yet to serve.

OFFICERS OF STUDENT BRANCHES

INSTITUTION	DATE AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	CORRESPONDING SECRETARY
Stevens Inst. of Tech. Hoboken, N. J..	Dec. 4, 1908	Alex C. Humphreys	A. E. Bauhan	A. D. Karr
Cornell University Ithaca, N. Y.	Dec. 4, 1908	R. C. Carpenter	F. E. Yoakem	D. S. Wegg, Jr.
Armour Inst. of Tech. Chicago, Ill.	Mar. 9, 1909	G. F. Gebhardt	A. J. Beerbaum	P. L. Keachie
Leland Stanford Jr. Univ., Palo Alto, Cal.	Mar. 9, 1909	W. F. Durand	C. H. Shattuc k	C. W. Scholefield
Polytechnic Institute Brooklyn, N. Y.	Mar. 9, 1909	W. D. Ennis	A. L. Palmer	R. C. Ennis
Purdue University Lafayette, Ind.	Mar. 9, 1909	L. V. Ludy	L. Jones	H. E. Sproull
University of Kansas Lawrence, Kans.	Mar. 9, 1909	P. F. Walker	V. H. Hilford	L. L. Browne
New York University New York	Nov. 9, 1909	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois Urbana, Ill.	Nov. 9, 1909	W. F. M. Goss	F. L. Schlink	E. J. Hasselquist
Penna. State College State College, Pa.	Nov. 9, 1909	J. P. Jackson	J. A. Kinney	H. S. Rodgers
Columbia University New York	Nov. 9, 1909	Chas. E. Lucke	N. E. Hendrickson	W. E. Ruprecht
Mass. Inst. of Tech. Boston, Mass.	Nov. 9, 1909	Gaetano Lanza	J. A. Noyes	R. M. Ferry
Univ. of Cincinnati Cincinnati, O.	Nov. 9, 1909	J. T. Faig	C. J. Malone	J. H. Schneider
Univ. of Wisconsin Madison, Wis.	Nov. 9, 1909	H. J. B. Thorkelson	F. B. Sheriff	L. F. Garlock
Univ. of Missouri Columbia, Mo.	Dec. 7, 1909	H. Wade Hibbard	G. D. Mitchell	P. A. Tanner
Univ. of Nebraska Lincoln, Neb.	Dec. 7, 1909	C. R. Richards	W. O. Forman	C. A. Bennett
Univ. of Maine Orono, Me.	Feb. 8, 1910	Arthur C. Jewett	A. H. Blaisdell	W. B. Emerson
Univ. of Arkansas Fayetteville, Ark.	Apr. 12, 1910	B. N. Wilson	W. Q. Williams	H. W. Barton
Yale University New Haven, Conn..	Oct. 11, 1910	L. P. Breckenridge	F. M. Jones	W. St. C. Childs
Rensselaer Poly. Inst. Troy, N. Y.	Dec. 9, 1910	A. M. Greene, Jr.	G. K. Palsgrove	H. J. Parthesius
State Univ. of Ky. Lexington, Ky.	Jan. 10, 1911	F. P. Anderson	J. W. Cary	J. T. Lowe
Ohio State University Columbus, O.	Jan. 10, 1911	E. A. Hitchcock	H. T. Lang.	W. J. Assel
Washington University St. Louis, Mo.	Mar. 10, 1911			F. E. Glasgow
Lehigh University South Bethlehem, Pa.	June 2, 1911			

SUMMARY OF MEMBERSHIP

United States

Alabama	21	Nebraska	6
Alaska	2	Nevada	3
Arizona	6	New Hampshire	16
Arkansas	3	New Jersey	206
California	99	New Mexico	2
Colorado	34	New York	1117
Connecticut	147	North Carolina	12
Delaware	20	North Dakota	1
District of Columbia	34	Ohio	274
Florida	3	Oklahoma	2
Georgia	21	Oregon	8
Hawaii	4	Pennsylvania	475
Idaho	2	Philippine Islands	1
Illinois	256	Porto Rico	4
Indiana	66	Rhode Island	69
Iowa	14	South Carolina	4
Kansas	11	Tennessee	15
Kentucky	5	Texas	28
Louisiana	28	Utah	6
Maine	17	Vermont	15
Maryland	31	Virginia	32
Massachusetts	394	Washington	18
Michigan	136	West Virginia	8
Minnesota	20	Wisconsin	102
Mississippi	2	Wyoming	2
Missouri	67		
Montana	9	Total	3878

Foreign Countries

Africa	14	Hungary	1
Australia	8	India	3
Austria	1	Italy	3
Belgium	5	Japan	6
Canada	58	Mexico	12
Central America	8	Norway	1
China	3	Russia	4
Cuba	6	Scotland	3
England	54	South America	15
Federated Malay States	1	Sweden	4
Finland	2	Switzerland	2
France	7		
Germany	13		
Holland	1	Total	235

SUMMARY OF MEMBERSHIP

By Residence

December 31, 1911

Membership in United States	3878
Membership in foreign countries	235
Present address unknown	2
Total membership	4115

By Grade

Honorary Members	16
Members	2774
Associates	374
Juniors	951
Total membership	4115

GAS POWER SECTION

SUMMARY OF MEMBERSHIP

United States

Alabama	4	New Hampshire	1
Arkansas	1	New Jersey	13
California	9	New York	143
Connecticut	7	Ohio	25
Delaware	1	Oklahoma	1
District of Columbia	6	Oregon	2
Georgia	1	Pennsylvania	34
Illinois	24	Rhode Island	6
Indiana	6	South Dakota	1
Iowa	1	Texas	1
Kansas	1	Vermont	1
Maine	3	Virginia	1
Maryland	1	Washington	1
Massachusetts	23	West Virginia	1
Michigan	13	Wisconsin	20
Minnesota	6		
Missouri	6	Total	365
Nebraska	1		

Foreign Countries

Belgium	1	Mexico	1
Canada	2	South America	3
England	1		
Germany	2	Total	10

By Residence

Membership in United States	365
Membership in foreign countries	10
Address unknown	1
Total membership	376

By Grade

Members of the Society	253
Affiliates	123
Total	376

STUDENT BRANCHES

Armour Institute of Technology	24
Columbia University	50
Cornell University	211
Lehigh University	8
Leland Stanford Jr. University	15
Massachusetts Institute of Technology	44
Ohio State University	34
Pennsylvania State College	40
Polytechnic Institute of Brooklyn	15
Purdue University	19
Rensselaer Polytechnic Institute	30
State University of Kentucky	50
Stevens Institute of Technology	66
University of Arkansas	1
University of Cincinnati	38
University of Illinois	65
University of Kansas	17
University of Maine	1
University of Missouri	29
University of Nebraska	16
University of Wisconsin	22
Washington University	16
Yale University	57
Total	868

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TRANSACTIONS

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

VOLUME 33—1911

THIS volume covers the activities of The American Society of Mechanical Engineers for the year 1911, the thirty-second in its history, including papers and proceedings of its various gatherings. A record of the work of the past year is given in the Annual Report of the Council, presented at the Annual Meeting of 1911.

At the Annual Meeting of 1910, an account of which appears in Volume 32 of Transactions, the newly-elected president, Edward Daniel Meier, was introduced.

EDWARD DANIEL MEIER

Edward Daniel Meier, president and chief engineer of the Heine Safety Boiler Company, was born in St. Louis, Mo., May 30, 1841. At the close of a scientific course at Washington University, St. Louis, in 1858, he spent four years in Germany at the Royal Polytechnic College in Hanover, this being followed by an apprenticeship at Mason's Locomotive Works, Taunton, N. J. In 1863 he enlisted in the Grey Reserves, the Thirty-Second Pennsylvania, which was attached to the army of the Potomac until after the Battle of Gettysburg. He subsequently served in the Second Massachusetts Battery, also in the United States Engineer Corps, and finally became lieutenant in the First Louisiana Cavalry, seeing much active service, and on May 30, 1865, receiving the surrender of Lieutenant-General John B. Hood and staff.

At the close of the war he entered the Rogers Locomotive Works at Paterson, N. J., remaining one year. From 1867 to 1870 he was associated with the Kansas Pacific Railway, first as assistant

superintendent of machinery, keeping open its Western communications when the bridges were swept away, designing, building and operating a mill for sawing, planing and turning the soft magnesian limestones by machinery, designing machine and car shops, etc., and subsequently becoming superintendent of machinery. He resigned to become chief engineer of the Illinois Patent Coke Company, leaving there in 1872 to assume the secretaryship of the Meier Iron Company and to build its blast furnaces. From 1873 to 1875 he directed the machinery department of the St. Louis Interstate Fair. During this time he became actively interested in the St. Louis cotton industry and was associated with the St. Louis Cotton Factory and with the Peper Hydraulic Cotton Press, for both of whom he designed machinery for compressing cotton. In 1884 he organized the Heine Safety Boiler Company for the development in the United States of the water-tube boiler of that name, and has been its president and chief engineer ever since. He has also been responsible for the introduction of the Diesel motor into the United States and until 1908 was engineer-in-chief and treasurer of the American Diesel Engine Company.

Colonel Meier was lieutenant-colonel and later colonel of the First Regiment of the Missouri National Guard, serving about ten years, and is a member of the Grand Army of the Republic and of the Loyal Legion. He has been active in a number of professional organizations, serving in 1881-1884 as treasurer of the St. Louis Engineers Club, in 1889-1890 as its president and as secretary of the American Boiler Manufacturers Association. It was in the latter capacity that he drew up the Uniform American Boiler Specifications of 1898. He has been president of that organization and also of the Machinery and Metal Trades Association since 1908.

In the Society he has been active on many committees, was one of its managers from 1895 to 1898, and has twice been elected vice-president, serving his first term from 1898 to 1900, and beginning the second in 1910, but resigning in the same year to assume the presidency.

REPORTS OF COUNCIL AND COMMITTEES

ANNUAL REPORT OF THE COUNCIL, 1911

The Council presents the following report of Society affairs during the year ending December 1911:

J. Sellers Bancroft, elected Manager in December 1910, resigned as a member of the Council and H. G. Stott was elected by the Council to fill the unexpired term. Alex. C. Humphreys was elected Vice-President to fill the unexpired term of E. D. Meier.

The Council also announces with deep regret the death of Charles Wallace Hunt, Past-President, and desires to spread on the permanent record the resolution taken at its meeting of April 10, 1911:

Charles Wallace Hunt, Past-President of The American Society of Mechanical Engineers, passed away on March 27, 1911, after twenty-six years of active and effective service as a member, officer, President and Past-President of the Society; and

Whereas the Council in session after this blow has befallen the Society is desirous to record its sorrow and sense of loss and its sympathy with those to whom this bereavement has come so sorely close;

Be it Resolved that the Honorary Secretary be requested to prepare a memorial notice to be published in The Journal with a portrait of Mr. Hunt; and be it further resolved that this minute commemorative of the great and notable services of Mr. Hunt to the Society be similarly published, and a copy of both be transmitted by the Secretary to Mrs. Hunt and the family with expressions of the keen and profound sympathy of Mr. Hunt's friends in the burden in which they are called on to bear.

The following have been appointed representatives of the Society on the various committees on which Mr. Hunt was serving: Committee on Society History, F. R. Hutton; John Fritz Medal Board of Award, W. F. M. Goss; Trustee United Engineering Society, Alex. C. Humphreys; Constitution and By-Laws, H. G. Stott.

MEMBERSHIP

The following table shows the changes in membership for the fiscal year October 1, 1910, to October 1, 1911, in distinction from the administration year, to which the other statements apply:

Grade	Oct. 1, 1910	Losses				Additions		Net Increase	Oct. 1, 1911
		Transfer	Resignation	Lapsed	Death	Transfer	Election		
Honorary	15					..	1	1	16
Members..	2609		32	15	44	39	145	93 (decrease)	2702
Associates....	393	15	4	13	1	..	10	23	370
Juniors	882	24	19	22	5	..	107	37	919
Total	3899	39	55	50	50	39	263	108	4007
Affiliates of Gas Power Section.	145	.		.	..	..	...	11 (decrease)	134
Affiliates of Student Branches	512	..	..	..	..	..	...	123	635

The Council has elected to Honorary Membership, J. A. F. Aspinall of Liverpool, England, Past-President of the Institution of Mechanical Engineers, for his original research work, exhaustive tests on train resistance, electrification of railways, and valuable contributions to the development of the locomotive.

On the Spring and Fall Ballots 324 new members were elected.

On the Council of The American Association for the Advancement of Science, Alex. C. Humphreys and H. G. Reist, were appointed.

GIFTS

Beside the large number of gifts of books to the Library, recorded in The Journal from time to time and reported by the Library Committee, the Society has been the recipient through the kindness of James M. Dodge, Past-President, of an autograph letter of James Watt, together with an engraving of a portrait of Watt by Sir William Beechy, R.A.

On the occasion of the seventieth birthday of Rear-Admiral Melville, Honorary Member and Past-President, a number of friends presented him with a silver plaque and engrossed resolutions of congratulation which the Admiral has since placed in the keeping of the Society.

Through the efforts of the friends of the late Charles Wallace Hunt, there has been placed in the hands of the Library Committee a memorial fund which is to be used for the purchase of books dealing with the transportation of materials.

STUDENT BRANCHES

There has been added to the list of Student Branches the State University of Kentucky, Ohio State University, Washington University, and Lehigh University. There is now a student enrollment of 841.

The Council has approved during the past year a form of insignia for the student branch members which has been adopted with much satisfaction by several branches.

AFFILIATED SOCIETIES

During the year a new relation has been inaugurated which is now in operation with the Providence Association of Mechanical Engineers, whereby the affiliated society secures the advantages of papers presented before the Society, and subscription to The Journal at the affiliate rate, without giving up in any way its independence. The rules governing such relations are incorporated elsewhere in this report under amendments.

MEETINGS OF THE SOCIETY

The Council has taken under its direct supervision the conduct of meetings of the Society other than the regular Spring and Annual meetings. By this arrangement the members of the Society in any city may organize local meetings, which will be open to all members of the Society and such other persons as the local committee may desire to be present. The local committee, consisting of five members of the Society nominated by the local membership and appointed by the Council, has full responsibility for the conduct of such meetings, and authority to secure professional papers of local or general interest, or institute topical discussions. Such papers as the general Committee on Meetings may select may also be presented at either the Spring or Annual Meetings of the Society.

The Society is now conducting meetings in Boston, New Haven, New York, Philadelphia, Providence, and San Francisco.

The precedent has been set during the past year of deciding at the Spring Meeting, the time and place of the next Spring Meeting.

LAND DEBT

Early in January the matter of paying off the Society's share of the mortgage on the land occupied by the Engineering Societies Building was considered by the Council and a special committee appointed,

consisting of Charles Wallace Hunt, Chairman, R. M. Dixon, and Waldo H. Marshall.

A plan was outlined by Mr. Hunt providing for the issuance of certificates of indebtedness, paying interest at 4 per cent per annum, payable semi-annually. This was approved and at the November meeting of the Council Messrs. Dixon and Marshall reported the complete success of the issue, the total subscriptions amounting to $85,800, of which $81,500 has been paid, and our share of the mortgage indebtedness thus cancelled. These certificates will be retired at the rate of $6000 or more annually.

JOHN FRITZ AUTOBIOGRAPHY

The Society has created the precedent during the year of publishing uniformly with the Transactions, an autobiography of its Past-President and Honorary Member John Fritz.

It is the intention from time to time as opportunity offers, to issue such publications as will enhance the interest of the engineering world in the men of attainments in the profession.

INTRODUCTION CARD

The form of identification card which has been inaugurated, combines the features of the card which has been issued previously and of a card of admission and identification for the Library.

For years the Society has exchanged courtesies with many of the engineering societies throughout the United States but it is doubtful if our members are aware of the fact. This has now been made apparent by giving on the reverse of the identification card a list of all such societies and libraries.

While the library is free and open to the public, it is often desirable that the member have access to the books themselves which are grouped according to subject, and this special privilege can be accorded only to persons known to this library administration. The identification card thus serves this double purpose.

AMENDMENTS TO THE CONSTITUTION, BY-LAWS AND RULES

The Council would report the following amendments to the Constitution, By-Laws and Rules during the past year:

B 13 The voting for the election of officers shall close at ten o'clock in the forenoon on the first Tuesday in December in each year: The Tellers shall not

receive any ballot after the stated time for the closure of the voting. The Tellers of Election shall first open and destroy the outer envelopes and shall then open the inner ones, canvass the ballots and certify the result to the President or Presiding Officer at the first session of the current meeting of the Society. The Presiding Officer shall then announce the candidates having the greatest number of votes for their respective offices and declare them elected for the ensuing year.

B 18 The Council at any meeting may, in its discretion, permanently remit the dues of any full Member of the Society who has been paying dues for thirty consecutive years or who shall have reached the age of seventy years, after having paid dues for twenty-five consecutive years, provided that notice of such proposed action shall have been given at a previous meeting of the Council and the Committee on Membership shall have concurred in recommending that this action be taken. The Council may in its discretion, restore to membership any person dropped from the rolls for non-payment of dues, or otherwise, upon such terms and conditions as it may at the time deem best for the interests of the Society.

B 20 The Council shall institute a monthly publication to be called "The Journal" which shall be under the management of the Secretary, who shall act under the general supervision of the Publication Committee, subject to approval by the Council as to the policy thereof and the expenditures therefor. The annual subscription price of The Journal to each member is two dollars and is included in the annual dues of such member.

B 23 The Committee on Meetings shall consist of five persons who may be members of any grade. The term of office of one member of the Committee shall expire at the end of each Annual Meeting. It shall be the duty of the Committee to procure professional papers, to pass upon their suitability for presentation, and to suggest topical subjects for discussion at the annual and semi-annual meetings. The Committee may refer any paper presented to the Society to a person or persons, especially qualified by theoretical knowledge or practical experience, for their suggestions or opinions as to the suitability of the paper for presentation. Papers from non-members shall not be accepted except by unanimous vote of the Committee.

The Committee shall arrange the program, and shall have general charge of the entertainments to be provided for the members and guests for the above meetings. It shall prohibit the distribution or exhibition at the headquarters or at the places of the above meetings of all advertising circulars, pamphlets or samples of commercial apparatus or machinery. At the end of each fiscal year, the Committee shall deliver to the Secretary for presentation to the Council, a detailed report of its work.

B 29 A Nominating Committee of five members, not members of the Council, shall be appointed by the President within three months after he assumes office. It shall be the duty of this Committee to send to the Secretary, on or before September 15, the names of consenting nominees for the elective offices next falling vacant under the Constitution. Upon the request of any Member or Associate, the Secretary shall furnish to the applicant the names of such nominees. The names of the nominees proposed by the Committee shall be published in the next issue of The Journal.

R 4 Papers accepted by the Committee on Meetings shall be classified as follows:

a Papers valuable for record, but not suitable for presentation at a meeting, which will be referred to the Publication Committee with the recommendations of the Committee on Meetings.

b Papers requiring either advance publication or printing and distribution in pamphlet form, either in full or in abstract for discussion at a meeting.

Papers of this class the Committee on Meetings may either turn over at once to the Publication Committee for advance publication in The Journal if approved by that Committee; or it may have printed in pamphlet form a sufficient number of copies for distribution at a meeting, for mailing in advance to those who may discuss the paper and to others who may desire copies because of publication of the title or an advance abstract of the paper in The Journal.

c Papers which require discussion at a meeting to determine their value and general interest.

Papers of this class may be accepted by the Committee on Meetings, but should not be put in type until they have been read and discussed.

The Committee on Meetings may in its discretion select papers for the annual or semi-annual meetings which have already been presented and discussed before local meetings of the Society.

A member may by letter signify his intention to discuss any of the papers, and unless otherwise directed by the Presiding Officer priority in debate shall be given in the order of the receipt by the Secretary of such notification.

R 36 Any regularly organized group of engineers or engineering society, which by constitution, by-laws and practice, is in accord with the traditions and aims of The American Society of Mechanical Engineers, may secure affiliation on approval by the governing boards of each Society.

R 37 The term "affiliated with The American Society of Mechanical Engineers" shall not be used officially or by individual members until specific approval has been secured by each governing board, nor shall such term be used by either after the relation of affiliation has terminated.

R 38 An affiliated society shall have independence, autonomy, and self-control under its own constitution and by-laws, unimpaired by its relation with The American Society of Mechanical Engineers.

R 39 The American Society of Mechanical Engineers shall not be responsible for any act of an affiliated society.

R 40 Printing for an affiliated society will be done at cost; this applies also to the printing in pamphlet form of advance copies of all papers presented before The American Society of Mechanical Engineers and such papers as may be desired for meetings of an affiliated society. The Society may print in The Journal any papers and discussions presented before an affiliated society, which the Publication Committee of The American Society of Mechanical Engineers may select.

R 41 The members of an affiliated society shall have the right to subscribe and receive The Journal on the same terms as other affiliates of The American Society of Mechanical Engineers.

R 42 The relation of affiliation with any organization may be terminated by either party upon sixty days notice, given in writing by the governing body of either society, to that of the other.

Rule 29 has also been renumbered and is now Rule 18, and the numbering of Rules 18 to 28, inclusive, has been changed to Rules 19 to 29.

FINANCE COMMITTEE

The arduous and successful work of the Finance Committee is deserving of the appreciation of every member of the Society. In order that the Society may be successful in its various activities for the benefit of the profession, it is also essential that its management be financially successful. That this is conspicuously the case will be shown by the perusal of the financial reports of the Society from year to year.

Under the leadership of Messrs. Dixon and Marshall of the Finance Committee, following the plan proposed by the late Chas. Wallace Hunt, the four per cent certificates of indebtedness to take up the Society's share of the mortgage on the building of the Engineering Societies, have been over-subscribed, and this share, together with all interest to date, has been paid off.

As an accompanying feature, the Society has pledged itself to retire at least six thousand dollars of these certificates each year, and has determined that this shall be made up mainly from the initiation fees.

In order that the above amount may be annually secured without reduction of the Society's financial resources for its routine activities, it is essential that the usual rate of increase of membership be maintained. It would be highly desirable if it were increased, thus extending the usefulness of the Society and its influence for good. The membership is therefore urged by the Finance Committee to undertake personally to secure for the Society the interest and membership of the leading engineers in all branches of the profession.

HOUSE COMMITTEE

The Council gratefully acknowledges the continuous work of the House Committee, which has increased the attractiveness of the rooms of the Society. Under its painstaking attention the rooms are made to look homelike and inviting.

LIBRARY COMMITTEE

The Library Committee have had a beautiful bookplate prepared.

The Council directs the attention of the membership to the permanent good which the Library Committee is doing by its painstaking

development of the resources of the Library and by its provision for complete researches under the direction of Mr. W. P. Cutter, a most competent librarian, with a staff of assistants. Thus is the Society being made available to members at a distance, which is the always present aim of the Council.

COMMITTEE ON MEETINGS

The Committee on Meetings have probably held more meetings than any other committee of the Society, and as a result of their faithful attention to their duties, the activities of the Society have been extraordinarily increased, extending the benefits to the members at a distance more than has ever been done in the history of the Society.

Meetings in several cities are now held periodically and subcommittees with especial relation to the several branches of engineering are being appointed.

PUBLICATION COMMITTEE

Too much praise cannot be given to the Publication Committee, who have, through their enterprise, been able to undertake a most intelligent and helpful form of advertising, not only of benefit to the members, but thereby bringing in sufficient income so that The Journal is not only self-supporting, but is undertaking features of incalculable value.

During the coming year, a topical index, supplemented by abstracts of some of the more important articles appearing in foreign languages, will be regularly published. Another important detail is the reduction of the price of The Journal to the public, making possible a wider distribution of the papers and discussions given at the meetings of the Society, thus adding to its prestige and benefiting the profession as a whole.

The Council desiring to show its appreciation of the work of this Committee during the past year, at its meeting November 20, passed the following resolution.

That this Council desires to express its high appreciation of the work of the Publication Committee in maintaining the high character of the Society's publications, and adding so largely to the Society's income during the past year.

MEMBERSHIP COMMITTEE

The character of the membership is maintained by the scrupulous care of the Membership Committee, and acknowledgment is enthusiastically given them for their pains.

Out of 401 applications, 368 have been favorably considered.

The work of the Committee would be greatly assisted if the membership would be careful to assist applicants to make their applications complete, particularly with respect to stating experience and duties performed, rather than positions held; also to advise applicants that references must have personal knowledge of these services.

The attention of the membership is particularly called to the dignity of the Associate grade, the age requirement of which is the same as for the grade of Member. Men of affairs associated with engineers are not only welcome in the Society, in the Associate grade, but their support and their interest is highly valued.

RESEARCH COMMITTEE

The Research Committee have been directing the accumulation of material, carrying out the plan of work of their committee, and have also appointed subcommittees to consider in detail special subjects, the general idea being to serve as a clearing house for information, rather than to direct or perform actual researches.

The Committee are securing a list of all the laboratories in which researches are being made, or are capable of being made, together with reports of the progress of each research. This will be published later, so that duplication of effort through the United States will not be possible, except by intention. This service also includes the notifying of all persons desiring suggestions as to what researches may advisedly be undertaken in any particular field.

PUBLIC RELATIONS COMMITTEE

The Public Relations Committee, with the enthusiastic interest of the President, has done a signal service to the profession as a whole by attempting to prevent ill-advised legislation in the State of New York, which would require all engineers to take out licenses. It is thought that by preventing such legislation in one State, particularly New York, that this type of legislation will not be favored in other States.

REPORT OF THE RESEARCH COMMITTEE

The activities of the Research Committee during the past year are not entirely revealed by the tangible results which have been accomplished. All that has been done must be regarded as preliminary to actual work.

The Secretary's office, under the direction of the Committee, has improved and extended its directory of the men who, as mechanical engineers, are engaged in work of scientific research in this country. There is also a directory of the laboratories in which these men are working. In the process of perfecting this record, an inquiry was made of 88 technical schools and universities, 63 of which number have reported. The list includes 5 physical laboratories, 15 chemical laboratories, 5 metallurgical laboratories, 33 electrical laboratories, 1 laboratory for the study of illumination and 54 dealing with problems of interest to the Committee. Of these 54 laboratories, 47 are especially equipped for steam engineering, 4 for gas engines, 33 for hydraulics, 26 for testing materials, 21 for refrigeration, and 6 for fuels. A small number are prepared for work in compressed air, for a study of oils, for sugar-house engineering, for the testing of road materials, for railway engineering, and for heating and ventilating. This information constitutes a permanent record in the file of the office of the Secretary and is available to all members.

There have been two formal meetings of the Committee, one in New York on December 6, 1910, and the other in Pittsburgh on June 1, 1911.

One of the results of the December meeting was a decision on the part of the Committee to ask authority of the Council to appoint sub-committees, which authority was duly given.

At the June meeting some progress was made in the development of plans for the sub-committees. The present status of this matter is as follows:

A Sub-Committee on Safety Valve Investigation, which is to deal with the whole question involving the action and the efficiency of safety valves, is in process of formation under the chairmanship of R. C. Carpenter, a member of the Research Committee. The personnel of this sub-committee cannot at this time be reported.

It has been proposed to have a Sub-Committee on Researches in Mechanical Properties of Materials Used in Electrical Engineering,

and R. D. Mershon, a member of the Research Committee, has been asked to act as Chairman.

The advisability of having a Sub-Committee on Fuel Utilization was considered, and it was voted that the Gas Power Section be advised that the Research Committee will depend upon it to cover this field. The Gas Power Section has been so advised.

It was agreed that there should be a Sub-Committee on Pneumatics and Aeronautics, but such a sub-committee has not yet been formed.

Your Committee desires in this connection to give some expression of the service performed as a member of the Committee by James Christie, whose death has occurred during the past year. Mr. Christie was among those who were faithful in attending Committee meetings and who gave to the deliberations of the Committee his best attention. His successor upon the Committee has not yet been appointed.

There has been formed a Sub-Committee on Steam Devices, the personnel of which is as follows: R. H. Rice, Member of the General Research Committee, Chairman; C. J. Bacon; E. J. Berg; W. D. Ennis; L. S. Marks; Max Patitz. They report concerning the present state of knowledge of the laws governing the transmission of heat through metallic tubes from gases and liquids to gases and liquids, that it is now rather definitely established that in the general case, the heat transfer takes place in at least three separate operations:

a The transfer from the warmer fluid to the initial surface of the tube.

b The heat passage through the tube wall from the warmer surface to the cooler surface of the tube.

c The transfer from the secondary tube wall to the cooler fluid.

This has been expressed by the general equation of Peclet

$$\frac{1}{k}=\frac{1}{c}+\frac{1}{q}+\frac{1}{r}$$

where

k = the coefficient of heat transmission.

q depends on the thermal conductivity of the tube which is known quite accurately and equals $\frac{K_t}{D}$ where D = thickness of tube; K_t = a constant depending on the material of which the tube is made.

r (following Osborne Reynolds) $=\frac{d_w V_w}{B}$; or (following Josse) =

$510\sqrt{V_w}$; where d_w = density of the cooling fluid, V_w = velocity of cooling fluid, and B = a constant.

c (following Pickworth and Nickolson) = $1.2V_s + 29400d_s - 306$, an empirical formula where V_s = velocity of the hotter fluid and d_s = its density.

As compared with c or r, q is always so large that in practical problems it can be neglected; in other words, the heat-transferring capacity of metallic tubes is always largely in excess of the heat which is brought in contact with the tube surface. For this reason, it is much more convenient to use a simpler expression, such as $K = C\sqrt{V_w}$.

Fourier stated the law as

$$dH = KSdz(T-t)$$

where S = surface, z = time, and T and t = temperature. When S and Z are unity, this expression becomes $H = K(T-t)$ and $(T-t)$ corresponds to θm in our modern expression. $H = N$, the total heat transferred, and $N = C\sqrt{V_w}\theta$ which is the form commonly used today.

There are many practical problems connected with the determination of the laws of heat transfer. Some of the more important are:

a The transfer from flue gas under the conditions existing in economizers. Here the transfer is of the order of 2 B.t.u. per sq. ft. per hr. per deg. difference of temperature.

b The transfer from flue gas to boiling water under boiler conditions (feedwater heated to steam temperature). Here the transfer is of the order of 10 B.t.u. per sq. ft. per hr. per deg. difference of temperature.

c The transfer from flue gases to steam or air under superheater or air heater conditions. This transfer is of the order of 4 B.t.u. per sq. ft. per hr. per deg. difference of temperature.

d The transfer from hot air to colder water under air cooler conditions. This is of the order of 4 B.t.u. per sq. ft. per hr. per deg. difference of temperature.

e The transfer from condensing steam to water under surface condensing conditions. This is of the order of 1500 B.t.u. per sq. ft. per hr. per deg. difference of temperature when no air is present, but falls off to 500 to 600 B.t.u. when an ordinary amount of air is present.

There seems to be an essential difference in the action when a condensible gas or vapor is the high temperature fluid. Here the vapor gives up its latent heat to the colder tube and contracts enormously in condensing, causing an exceedingly rapid flow normal to the tube. It may be that an analogous action takes place when the colder tube is exposed to the "bombardment" of radiant heat, as in the case of boiler tubes exposed to the rays from white hot surfaces or fuel or brickwork. Professor Bone in his lecture before the Royal Society last year stated that he had obtained in an experimental boiler an evaporation as high as 41 lb. of water per sq. ft. of surface, which corresponds to approximately 25 B.t.u. per sq. ft. per hr. per deg. difference. This was due to the radiant heat condition. Probably the best exposition of the problem where a permanent gas is the hotter fluid is given in Professor Dalby's report to the Institution of Mechanical Engineers, October 29, 1909, and his conclusions summed up on pages 25 to 28 show the exceedingly difficult nature of the problem and the necessarily costly means which must be used for its complete solution. The main factor is the determination of the temperature gradients in all parts of the boiler. That this is also true of the condenser conditions may be seen from the work of Smith and Josse and the paper read by Mr. Orrok before the Society last year.

The "mass flow" theory of Osborne Reynolds and Jordan seems to be another form of stating the fact that where more heat is present more is transmitted, and it is known that this is true up to the limit of heat transfer. Kreisinger and Ray have investigated this problem, with an experimental boiler using air heated electrically as the hotter fluid, and their results have been widely quoted. The work of Perry developed by Breckenridge in Bulletin 325 of the Geological Survey is also of great value in this connection.

Summarizing, the lines of work which will be most fruitful are as follows:

a A final determination of the radiant heat law (Stefan's law) with a separation from conduction and convection problems. It is probable that Stefan's law is nearly correct, but does the black body condition exist, and between what ranges of temperature? Bone's experiments seem to show that it may exist at temperatures below red heat. (See Bulletin No. 8, Department of Mines.)

It is known that in boiler conditions radiation is much more efficient than either conduction or convection. How much of the heat supply can be developed as radiant energy? Both Nicholson and

Bone claim that the boiler of the future will depend largely on radiation rather than conduction.

b More study of the phenomena of heat transfer in the absence of radiant energy (if this is possible). What is the effect of dust and soot in the gases? Do the dust particles act as black bodies and give out radiant energy? It has been stated that 98 per cent of the temperature drop took place in the passage of the heat from the gas to the tube surface. Is this statement correct, and if so, can the "gas film" be removed or made a better conductor?

c Determination of the temperature gradient under a variety of practical conditions, and with full size apparatus. Does the gas film give more trouble when inside a fire tube with axial flow of gases or when outside a water tube with a flow across the surface?

d A determination of the law of temperature rise in the water flowing in a condenser tube. The tube should be long and the thermometers numerous.

e A repetition of Smith's experiments with the addition of known amount of air and with the use of very sensitive thermometers. This series should include all commercial vacua, and the lower end of the steam table should be recalculated with much smaller intervals for use in interpreting the results.

f The expressions, radiant heat, latent heat, etc., have been used. Sensible heat is defined as molecular motion. Radiant heat has been defined as ether waves. When latent heat is set free as in a condenser, is it molecular or ethereal in its constitution? In chemistry, when an element is set free from combination, there is what is known as the nascent state. Does an analogous action take place when heat is set free from the latent state?

Some of these questions should perhaps be answered by the physicist rather than the engineer, but seem to the Committee to be true engineering problems to be studied through their applications if a complete, reasonable and satisfying theory is to be obtained.

Respectfully submitted,

W. F. M. Goss, *Chairman*
R. C. Carpenter
R. D. Mershon
R. H. Rice

Research Committee

REPORT OF COMMITTEE ON IDENTIFICATION OF POWER HOUSE PIPING

a In the main engine rooms of plants which are well lighted, and where the functions of the exposed pipes are obvious, all pipes shall be painted to conform to the color scheme of the room; and if it is desirable to distinguish pipe systems, colors shall be used only on flanges and on valve fitting flanges.

b In all other parts of the plant, such as boiler house, basements, etc., all pipes (exclusive of valves, flanges and fittings), except the fire system, shall be painted black, or some other single, plain, durable, inexpensive color.

c All fire lines (suction and discharge), including pipe lines, valve flanges and fittings, shall be painted red throughout.

d The edges of all flanges, fittings or valve flanges on pipe lines larger than 4 in. inside diameter, and the entire fittings, valves and flanges on lines 4 in. inside diameter and smaller, shall be painted the following distinguishing colors, numbered 1 to 12, inclusive:

DISTINGUISHING COLORS TO BE USED ON VALVES, FLANGES AND FITTINGS ONLY

Steam division	*a*	High pressure—white
	b	Exhaust system—buff
Water division	*c*	Fresh water, low pressure—blue
	d	Fresh water, high pressure boiler feed lines—blue and white
	e	Salt water piping—green
Oil division	*f*	Delivery and discharge—brass or bronze yellow
Pneumatic division	*g*	All pipes—gray
Gas division	*h*	City lighting service—aluminum
	i	Gas engine service—black, red flanges
Fuel oil division	*j*	All piping—black
Refrigerating system	*k*	White and green stripes alternately on flanges and fittings, body of pipe being black
Electric lines and feeders	*l*	Black and red stripes alternately on flanges and fittings, body of pipe being black

Respectfully submitted,

H. G. STOTT, *Chairman*
F. R. HUTTON
I. E. MOULTROP
H. P. NORTON
J. T. WHITTLESEY

Committee on Identification of Power House Piping

No. 1306

MEETINGS JANUARY–JUNE

MEETINGS PREVIOUS TO THE SPRING MEETING

NEW YORK, JANUARY 10

The Mechanical Handling of Freight: Paper by Samuel B. Fowler, consulting engineer, of Boston. This paper treats of the lack of adequate terminal facilities, increase of net income and lower freight rates, and the possibility of increasing these facilities and solving the present problems by the substitution of mechanical devices for manual labor and hand trucks. Published in The Journal, January 1911. For the discussion on this paper see The Journal, July 1912.

PHILADELPHIA, JANUARY 28

Discussion of project for holding meetings of the Society in Philadelphia on the plan of those already in progress in other cities.

BOSTON, JANUARY 31

Engineers' dinner, attended by more than 400 representatives of different branches of the profession, at the Hotel Somerset, under the charge of the Boston section of the American Institute of Electrical Engineers, the Boston Society of Civil Engineers and The American Society of Mechanical Engineers coöperating. John F. Vaughan, Chairman of the Boston section of the American Institute of Electrical Engineers presided, and the speaker of the evening was Prof. Elihu Thompson, Mem. A. I. E. E., of Lynn, Mass. Addresses were also made by J. J. Carty, Mem. A. I. E. E., chief engineer of the American Telephone and Telegraph Company, who represented the American Institute of Electrical Engineers in the absence of President Jackson; Col. E. D. Meier, President Am. Soc.M.E.; and Henry F. Bryant, President of the Boston Society of Civil Engineers. Prof. Ira N. Hollis, Mem. Am. Soc. M. E., spoke of the status of plans for an engineers' building in Boston. An account of this meeting was given in The Journal for March 1911.

NEW YORK, FEBRUARY 14

The Mechanical Engineer and the Prevention of Accidents: Paper by John Calder. This discusses the nature and incidence of industrial injury, its prevalence and high rate in the United States in particular, and the present general desire for better conditions of safety. The chief causes of injury are analyzed by means of a study by the author of a large number of verified casualties, and practicable remedies are recommended. The part of the mechanical engineer in this matter is especially discussed.

The paper appears in The Journal, February 1911; discussion published July 1911.

BOSTON, FEBRUARY 17

Economic Limitations to Aggregation of Power Systems: Paper by R. A. Philip, Assoc. A. I. E. E., of the Stone and Webster Engineering Corporation, Boston. Published in Proceedings of the American Institute of Electrical Engineers for February 1911.

NEW YORK, MARCH 10

Meeting with the American Institute of Electrical Engineers. Papers: Comments on Fixed Costs in Industrial Power Plants, and Notes on the Cost of Electrical Energy, by John C. Parker; and The Cost of Industrial Power, by A. E. Hibner, Assoc. A. I. E. E. Published in Proceedings of the Institute, March 1911.

SAN FRANCISCO, MARCH 10

Second meeting for consideration of Pacific Coast Practice in the Use of Crude Petroleum. Papers and discussion published in The Journal, August 1911, and in condensed form in this volume.

BOSTON, MARCH 17

Speed Regulation in Hydroelectric Plants: Paper by Wm. F. Uhl. Published in The Journal, February 1911. See also Volume 34 of Transactions.

ST. LOUIS, MARCH 25

Business meeting, addressed by Col. E. D. Meier, President Am. Soc. M. E.

NEW YORK, APRIL 11

The Economic Importance of the Farm Tractor: Paper by L. W. Ellis, traction plowing specialist of the M. Rumely Company, LaPorte, Ind. Published in The Journal, March 1911, and in this volume in abstract form.

BOSTON, APRIL 21

Oil Fuel for Steam Boilers: Paper by B. R. T. Collins. Published in the Journal, August 1911, and in this volume in abstract form.

PHILADELPHIA, APRIL 22

The Work of the United States Fuel Testing Station: Paper by J. C. Parker, giving curves plotted from the results published by the United States Fuel Testing Station.

PROVIDENCE, MAY 3

Engineers' dinner, for organization of meetings in Providence. Addressed by Edwin C. Bliss, President of the Providence Association of Mechanical Engineers; William Howard Paine, acting as toastmaster; Calvin W. Rice, Secretary Am.Soc.M.E.; Col. E. D. Meier, President Am.Soc.M.E.; F. R. Hutton, Honorary Secretary Am.Soc.M.E. An account of this meeting appears in The Journal for June 1911.

NEW YORK, MAY 9

Patent Law: Papers on the More Fundamental Principles of Patent Law, by Edwin J. Prindle; Letters Patent for Inventions, by D. Howard Haywood; the Patent Expert, by E. W. Marshall. The first of these papers appears in The Journal, July 1912, and the second in this volume. All three papers are available in pamphlet form

BOSTON, MAY 17

The Electric Motor in the World's work: Paper by Fred. M. Kimball, Assoc.A.I.E.E., West Lynn, Mass., briefly describing the development of the electric motor from the discovery of the fundamental principles early in the nineteenth century up to the present time.

PHILADELPHIA, JUNE 3

Continuation of meeting on Fuel Testing, held in April.

ST. LOUIS, JUNE 7

Engineers' dinner, under the auspices of the Engineers Club of St. Louis. J. D. von Maur, president of the Club, acted as toastmaster and spoke of the joint professional meetings in which the members of the American Society of Civil Engineers, the American Institute of Electrical Engineers, The American Society of Mechanical Engineers, the American Society of Engineering Contractors, and the Engineers Club had participated. M. L. Holman, Past-President Am.Soc.M.E. and of the Engineers Club; Col. E. D. Meier, President Am.Soc.M.E., C. M. Woodward, Past-President of the Engineers Club and founder of the first manual training school in the country, and Col. Ed. Devoy were among the speakers of the evening. An account of the meeting appears in The Journal for July 1911.

THE SPRING MEETING

The Spring Meeting was held at Pittsburgh, Pa., May 30 to June 2, 1911, with an attendance of 307 members and 353 guests. Most of the social gatherings were held at the Hotel Schenley, where the registration headquarters were maintained, but the business meetings and professional sessions were held in the lecture hall of the Carnegie Institute and in the Carnegie Technical Schools.

This meeting was an unusually strong combination of pleasant social features, interesting technical excursions and valuable professional sessions. The papers were of unusual interest, and in spite of many other attractions the professional sessions were all well attended.

PROGRAM

Tuesday Afternoon and Evening, May 30

Registration and informal reunion in the parlors of the Hotel Schenley.

Wednesday Morning, May 31

Business meeting. Reports of committees, tellers of election. New business.

SYMPOSIUM ON CEMENT MANUFACTURE

SOME PROBLEMS OF THE CEMENT INDUSTRY, Walter S. Landis.
EDISON ROLL CRUSHERS, W. H. Mason.
POWER AND HEAT DISTRIBUTION IN CEMENT MILLS, L. L. Griffiths.

Discussed by F. B. Gilbreth, R. K. Meade, W. R. Dunn, H. Struckmann, C. J. Reilly, G. P. Hemstreet, F. L. Schwenck, H. E. Brown, W. B. Ruggles, W. M. Kinney, Paul C. Van Zandt.

Wednesday Afternoon

Inspection trip to Universal Portland Cement Company Works at Universal and to the Westinghouse Electric & Manufacturing Company works and the Westinghouse Machine Company works at East Pittsburgh.

Inspection of the Foundry and Machine Company exhibit at Exposition Building.

Wednesday Evening

MACHINE SHOP SESSION

THE ASSEMBLY OF SMALL INTERCHANGEABLE PARTS, John Calder.
THE PROCESS OF ASSEMBLING A SMALL AND INTRICATE MACHINE, Halcolm Ellis.
QUANTITY MANUFACTURE OF SMALL PARTS, F. P. Cox.

Discussed by Hugo Diemer, E. Puchta, C. W. Ripsch, C. W. Johnson, W. J. Kaup, N. W. Perkins, Jr., A. C. Jackson.

MILLING CUTTERS AND THEIR EFFICIENCY, A. L. DeLeeuw.

Discussed by W. S. Huson, John Parker, A. F. Murray.

GAS POWER SECTION

TOPICAL DISCUSSION ON LARGE BLAST-FURNACE GAS-POWER PLANTS, A. E. Maccoun, R. H. Stevens, A. N. Diehl, A. L. Hoehr, H. J. Freyn, E. Friedlander.

The Work of the U. S. Bureau of Mines, S. B. Flagg and C. D. Smith.

No discussion.

Thursday Morning, June 1

Stresses in Tubes, R. T. Stewart.

No discussion.

The Purchase of Coal, D. T. Randall.

Discussed by R. C. Carpenter, W. F. M. Goss, E. D. Meier, C. W. Rice, C. W. Baker.

Energy and Pressure Drop in Compound Steam Turbines, F. E. Cardullo.

Discussed by W. H. Herschel, C. H. Peabody, W. D. Ennis.

The Pressure-Temperature Relations of Saturated Steam, L. S. Marks.

Presented by title only. No discussion.

A Pressure Recording Indicator for Punching Machinery, G. C. Anthony.

Discussed by R. C. Carpenter, Julian Kennedy.

Thursday Afternoon

Inspection trip to National Tube Company Works, McKeesport.

Thursday Evening

Reception and dance at Hotel Schenley.

Friday Morning, June 2

Steel Works Session

Commercial Application of the Turbine Turbo-Compressor, R. H. Rice.

Discussed by E. D. Dreyfus, J. E. Johnson, Jr., R. N. Ehrhart, C. G. de Laval, Julian Kennedy, Joseph Morgan, C. J. Bacon, H. J. Freyn.

RECIPROCATING BLAST-FURNACE BLOWING ENGINES, W. Trinks.

Discussed by E. T. Child, F. E. Cardullo.

POWER FORGING, WITH SPECIAL REFERENCE TO STEAM-HYDRAULIC FORGING PRESSES, Barthold Gerdau and George Mesta.

Discussed by J. I. Rogers and W. E. Hall.

Friday Afternoon

Inspection trip to Carnegie Steel Company Works, Duquesne, and Mesta Machine Company Works, West Homestead.

Friday Evening

Smoker and entertainment given by the Engineers Society of Western Pennsylvania in the Union Club.

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No. 1307

TOPICAL DISCUSSION ON FUEL OIL

THE PRODUCTION OF PETROLEUM ON THE PACIFIC COAST

By Arthur F. L. Bell, San Francisco, Cal.[1]

Non-Member

Attention was first drawn in the early 60's to the presence of petroleum in California by the discovery of numerous oil seepages in Los Angeles, Ventura, Santa Barbara and adjacent counties. Most of the early drilling was confined to Ventura County and until the discovery of the Los Angeles City field no other area exceeded about 40 acres. In 1882 the Los Angeles field came in with an approximate area of 400 acres. The oil territory traversed a narrow strip in the northwestern portion of the city, crossing hundreds of city lots, so that in consequence all lot owners became oil operators or leased their lots to others. The formation was so easily drilled that portable rigs were used and in many instances wells about 800 ft. in depth were completed in a week. As late as the year 1900, a well 2000 ft. in depth was considered exceptional and there were but a few in the State, whereas today we have wells of one mile or over in depth.

2 Such conditions resulted in an immediate over-production with a corresponding drop in price of oil from about $1.50 per bbl. previous to the discovery of the Los Angeles field, to from 15 to 30 cents per bbl. in Los Angeles in 1896. This low price caused every Southern California industry that required fuel in its operations to adopt oil on account of its cheapness and resultant saving. The real opening of California's oil industry may be said to date from the discovery of the Los Angeles field. In 1898 this field was on the wane and the price of oil rose to the dollar mark. The universal adoption of oil

[1] The Associated Oil Co.

In abstract form. Presented at San Francisco meetings, December 1910 and March 1911, of The American Society of Mechanical Engineers. A more complete account appears in The Journal, August 1911.

in Southern California had interested so many men in the industry, however, that new fields were sought, with the result that the Fullerton, Coalinga, Kern River, McKittrick, and Sunset fields were immediately exploited in the order named and by 1902 the price of oil again fell to 15 and 20 cents per bbl. in the Kern field. The production of the cheap Kern River oil forced its use over the entire Pacific Coast States as well as the Hawaiian Islands.

3 It should be understood that in the early development of the oil industry the discovery of oil was greatly a matter of chance. All of the early fields, including the Los Angeles City field, Kern River, Coalinga, Sunset, etc., were discovered by drilling close to oil seepages. The Summerland field was discovered by chance through digging a water well. The Kern field was discovered by a wood chopper who, having seen a small seepage on the bank of the Kern River, dug a well and with the proverbial good luck of the novice struck oil at about 75 ft.

4 Today the industry has become so well understood that, to extend the field, instead of the operator hugging the outcrop, he confidently locates his well far out in the valley or in the outlying ranges within the possible oil belt, in many cases miles from the nearest production. Where he has used fair judgment, based on the knowledge which we now have of the geological conditions of the oil measures, the chances of failure are greatly reduced. The present development of the oil industry is due, first, to the improvement in the art of drilling wells which has enabled us to reach successfully and economically depths of from 4000 to 5000 ft.; and, second, to knowledge of the oil-bearing formation, for which we are to a great extent indebted to the United States Geological Survey, which has for the last few years had a corps of geologists in the field mapping and cross-sectioning proven and possible oil territory. It is satisfactory to note that during the last year successful discoveries have been made in far outlying districts, reported as possible oil territory by the United States Geological Survey.

5 California oils are found in the tertiary formation in loose sandy beds. In many cases these sand strata are from 400 to 500 ft. thick, which allows for enormous storage of oil, whereas in the Eastern fields the older rocks have become so cemented and are so close-grained that there is no room for storing the oil, a difference which accounts for the small production of the Eastern wells compared with those of California.

6 In 1908, the United States Geological Survey reported 8450 sq. mi. (5,408,000 acres) of possible oil territory in the United States,

giving California 850 sq. mi. (544,000 acres), or about one-tenth of the total area, and also credit for one-half of the probable minimum oil production of the United States, or one-third of the probable maximum production. The contents of the probable oil lands of the United States in barrels of 42 gal. have also been estimated as follows:

	Minimum	Maximum		Minimum	Maximum
Appalachian..	2,000,000,000	5,000,000,000	Gulf.....	250,000,000	1,000,000,000
Lima-Indiana	1,000,000,000	3,000,000,000	California	5,000,000,000	8,500,000,000
Illinois.....	350,000,000	1,000,000,000	Minor.. .	1,000,000,000	5,000,000,000
Mid-continent	400,000,000	1,000,000,000	Total....	10,000,000,000	24,500,000,000

The following shows the average daily production per well in the various fields of the United States:

	Bbl.		Bbl.
Appalachian.....	1.73	Illinois.....	8.73
Lima-Indiana.....	2.74	Mid-continent.....	5.29
Colorado & Wyoming	8.35	Gulf.....	14.08
California	42.56		

7 These were conservative estimates when made two years ago, but the development in the industry has been so rapid that they will not answer for today. I have therefore made an estimate of what I consider the proven and prospective areas of today in California, which shows a total area of practically proven oil territory of 94,200 acres, with an estimated production of 4,319,000,000 bbl., and a probable extension of oil territory of 338,320 acres, with an estimated production of 13,258,000,000 bbl., making a grand total of proven territory of 432,520 acres and 17,577,000,000 bbl. The total consumption to date has been about 383,000,000 bbl., leaving an estimated future possible production of 17,194,000,000 bbl. Table 1 gives an itemized statement of the production by counties and fields.

8 In submitting the above estimate of extractable oil, which is about double the estimate of the United States Geological Survey, I can only say that it is impossible for any one to state definitely what our actual production will be. I have included as possible territory only what I consider, with our present knowledge, probable territory, amounting to about 700 sq. mi., but it is quite possible that a greater acreage will be developed which will bring the total up to or over the estimate of the United States Geological Survey of 850 sq. mi., which should again increase the quantity of prospective extractable oil.

9 In the computation I have made, I have assumed the interstices in the sand to be three-tenths of its bulk and that it is filled with

TABLE I ESTIMATED FUTURE OIL SUPPLY OF CALIFORNIA

County and Field	Proven Acreage	Thickness of Sand (Ft.)	Estimated Production (Bbl.)	Possible Acreage	Thickness of Sand (Ft.)	Estimated Production (Bbl.)	Totals By Fields: Acreage	Totals By Fields: Production (Bbl.)
Coalinga	29,000	50	1,128,000,000	21,440	50	834,000,000	50,440	1,962,000,000
Kettleman Hills				20,080	50	778,000,000	20,080	778,000,000
Total, Fresno County	29,000		1,128,000,000	41,520		1,612,000,000	70,520	2,740,000,000
Kettleman Hills				53,120	50	2,064,000,000	53,120	2,064,000,000
Total, Kings County				53,120		2,064,000,000	53,120	2,064,000,000
Lost Hills				46,080	50	1,800,000,000	46,080	1,800,000,000
Devils Den				14,720	50	596,000,000	14,720	596,000,000
Antelope Hills				16,000	50	622,000,000	16,000	622,000,000
McKittrick Front	2,000	50	85,000,000	6,400	50	250,000,000	8,800	380,000,000
McKittrick Front	400	150	45,000,000					
Midway Sunset	40,000	50	1,700,000,000	100,480	50	4,000,000,000	140,480	5,700,000,000
San Emidio				12,000	50	466,000,000	12,000	466,000,000
Kern River	6,000	150	590,000,000				6,000	590,000,000
Total, Kern County	48,400		2,420,000,000	195,680		7,734,000,000	244,080	10,154,000,000
Santa Maria	3,600	25	70,000,000				3,600	70,000,000
Lompoc	1,280	25	25,000,000	2,000	50	26,000,000	3,280	51,000,000
Cat Cañon	1,920	50	76,000,000	16,000	50	622,000,000	17,920	698,000,000
Total, Santa Barbara County	6,800		171,000,000	18,000		648,000,000	24,800	819,000,000
Southern California Fields	5,000	100	400,000,000	5,000	50	200,000,000	10,000	600,000,000
Miscellaneous	5,000	50	200,000,000	25,000	50	1,000,000,000	30,000	1,200,000,000
Total Proven Acreage	94,200							
Total Possible Acreage				338,320				
Estimated Production Proven Acreage			4,319,000,000					
Estimated Production Possible Acreage						13,258,000,000		
Final Grand Totals							432,520	17,577,000,000
Less Oil Produced to 1911								383,000,000
California's Estimated Future Supply								17,194,000,000

oil of which one-third would be held by capillary attraction to the sand, one-third as readily extractable oil, and the remaining one-third as oil which can not be obtained. This would admit of producing about ¾ gal. per cu. ft. of oil sand, or 750 bbl. per acre foot. To substantiate these figures, I will state that prior to 1900, the Alcatraz Asphalt Company in Santa Barbara County, mechanically extracted from the bituminous sands at its Carpenteria and Sisquoc refineries, 27 per cent and 25 per cent respectively of bitumen. These sands were exposed oil sands from which the bitumen or dried-out oil was being extracted.

10 In 1901, Bernard Bienenfeld, William Mulholland, and the writer, acting as commissioners for the purpose of appraising the value of the different properties in the Kern field that were to form the Associated Oil Company, determined that the Kern field oil sand did contain 32 per cent of void in the sand and when filled with oil, amounted to 2.4 gal. per cu. ft. It will therefore be seen that an estimate based on an extraction of ¾ gal. per cu. ft., or 750 bbl. per acre ft., should not be considered as unreasonable if only the correct thickness of productive sands is figured.

11 To determine the life of our field, taking the submitted figures as a basis, we would have, if we assumed the present production of 75,000,000 bbl. as the maximum consumption, 52½ years' supply for the proven territory, and for the proven and prospective territory, less consumption to date, about 230 years' supply. If our consumption is increased, as it naturally will be, the life of our fields must be reduced proportionally. Our consumption for 1910 will amount to about 63,000,000 bbl., as about 12,000,000 bbl. will have gone into storage this year, and at the end of 1910 we will have in storage about 30,000,000 bbl., or about five months' supply. The following is a list of the recorded production of California from 1875 to 1910, inclusive, and the average daily production per well:

	Bbl. produced	Average bbl. per well per day
1875	3,000	4.11
1880	40,552	15.83
1885	325,000	38.70
1890	307,360	15.88
1895	1,208,482	13.19
1900	4,324,484	9.15
1905	33,427,473	33.49
1910 (estimated)	75,000,000	42.10
Total	382,821,212	

COMPARATIVE EVAPORATIVE VALUES OF COAL AND OIL

By C. F. Wieland, San Francisco, Cal.

Member of the Society

12 In comparing the evaporative values of the two fuels, coal and oil, it is only fair to select a coal that is considered standard and that is largely used in evaporative tests. It would not be just to select a coal of the inferior value of our Western coals, and Pocahontas coal and Kern crude oil are therefore selected as a basis of comparison, having analyses as follows:

	Pocahontas Coal, Per Cent	Kern Crude Oil, 14.5 Gravity, Per Cent
Proximate Analysis		
Fixed carbon	73.30	.....
Volatile matter	17.61	.....
Moisture	0.49	0.15
Ash	8.60	.....
	100.00	
Ultimate Analysis		
Carbon	82.26	87.64
Hydrogen	3.89	10.48
Sulphur	0.49	1.02
Oxygen	4.12	0.08
Nitrogen	0.64	0.78
Ash	8.60	.....
Sludge	.....	.....
	100.00	100.00
Calorific B.t.u.	14,067	18,619

Coal data from Naval Liquid Fuel Board report, p. 10.

13 Assuming the complete combustion of the heat values in the two fuels as analyzed and knowing that 1 lb. of C requires 11.6 lb. air, 1 lb. of H requires 34.8 lb. air, and 1 lb. of S requires 4.3 lb. air, we have as the volume of air chemically required for the combustion of the coal and the oil the results shown in Table 2.

14 For our comparison we will make the following assumptions:

a Flue gas temperature is the same in both cases, viz., 500 deg.

b Moisture in the two fuels is the same.

c Air intake temperature (temperature of fireroom), 70 deg.

The flue temperature as placed at 500 deg. is probably the mean which obtains in practice; lower results have been obtained, but numerical value makes little difference in our comparison.

15 Moisture as shown in the selected coal is low and that in the selected oil is also low. In most oils it is more nearly an average of 0.5 per cent and for this reason we will consider that the moisture content is the same in each case.

TABLE 2 VOLUME OF CHEMICALLY REQUIRED AIR FOR COMBUSTION OF COAL AND OIL

	Weight of Air Required	
	Coal	Oil
Carbon		
Coal, 11.6 lb. air × 0.8226 lb. C	9.54	....
Oil, 11.6 lb. air × 0.8764 lb. C	....	10.16
Hydrogen		
Coal, 34.8 lb. air × 0.0389 lb. H	1.35	....
Oil, 34.8 lb. air × 0.104 lb. H	....	3.63
Sulphur		
Coal, 4.3 lb. air × 0.0049 lb. S	0.02	....
Oil, 4.3 lb. air × 0.010 lb. S	....	0.04
Lb. of air required	10 91	13.82
Weight of the combustibles (after deducting ash from coal)	0.91	1.00
Total weight which will be considered as dry chimney gas	11.82	14.82

16 The sensible heat carried away in the flue gases in the case of the coal is

$$0.24\ (500\text{–}70) \times 11.82 = 1219.8\ \text{B.t.u.}$$

In the oil

$$0.24\ (500\text{–}70) \times 14.82 = 1529.4\ \text{B.t.u.}$$

0.24 being the specific heat of the chimney gases.

17 In the combustion of hydrogen to water each pound of H results in 9 lb. of water, or in our coal the water formed will be

$$0.038 \times 9 = 0.34\ \text{lb.}$$

18 The heat loss both latent and sensible in the evaporation of this water is

$$0.34\ (142 + 966 + 288 \times 0.47) = 422.6\ \text{B.t.u.}$$

wherein 212 deg. − 70 deg. = 142, assuming the temperature of the water in the coal to be that of the fireroom; 966 being the latent heat of the formation of steam at 212 deg. and of expansion against the atmosphere; 500 deg. − 212 deg. = 288 being difference be-

tween steam at 212 deg. and flue temperature as assumed; and 0.47 being the specific heat of superheated steam at atmospheric pressure.[1]

19 In case of oil we have

$$0.104 \times 9 = 0.936 \text{ lb.}$$

Heat loss is

$$0.936\ (62 + 966 + 288 \times 0.47) = 1088.5 \text{ B.t.u.}$$

The loss in coal due to ash is 8.6 per cent of total = 1209.7 B.t.u. A further loss in oil combustion is the heat absorbed by the steam used in atomizing, being in superheating the steam from 212 deg. to 500 deg. fahr. This is

$$288 \times 0.47 = 135 \text{ B.t.u.}$$

Tabulating these losses, which for convenience we will call fixed losses, we have:

	Coal	Oil
Gases	1219.8 B.t.u.	1529.4 B.t.u.
Combustion of H	422.6 B.t.u.	1088.5 B.t.u.
Ash	1209.7 B.t.u.	Burner 135.0 B.t.u.
	2852.1 B.t.u.	2752.9 B.t.u.

While this calculation is not intended to convey any idea of the actual value of losses in the two fuels, it may however afford a comparison of what we may term fixed losses, within certain limits.

20 It is seen that the amount of air required for the combustion of coal is very much less than for oil. This is due principally to the very much greater hydrogen content in the oil, the carbon content being, roughly speaking, the same. Furthermore the combustion of the hydrogen requires a proportionately larger weight of air.

21 On the other hand the greatest loss in the coal is in the ash, as in the comparison of the fuels we must consider them as received. In the oil the loss due to non-combustible is practically negligible. The loss due to superheating the burner steam is also small.

22 In examining our figures a remarkable equality of the total losses is immediately apparent. To what extent this relation would hold for coals of different composition would have to be determined. Subtracting these losses from the calorific value of each fuel we have:

$$18{,}619 - 2752.9 = 15{,}866.1 \text{ B.t.u.}$$

$$14{,}067 - 2852.1 = 11{,}214.9 \text{ B.t.u.}$$

These last quantities represent the available heat left in fuel for evap-

[1] R. C. H. Heck. Thermal Properties of Superheated Steam, Trans. Am. Soc. M. E., vol. 30, p. 227.

oration. From this it would appear that the two fuels under examination had an evaporative value which is in the ratio of 11.2 for coal to 15.8 for oil. For coals equal in quality to that selected, the writer believes that this ratio will generally be correct, lower grade coals having correspondingly lower values. This ratio is also amply borne out in numerous actual tests.

23 It is shown in the foregoing that the theoretically required air supply is greater for oil than for coal. This means a greater volume of flue gases, hence a greater loss of heat. If we consider, however, those coals having a large amount of volatile matter, it is safe to say that the air supply for their complete combustion would be very much in excess of that for oil. Not knowing the composition of the volatile matter, we must necessarily figure with a certain indefinite quantity of excess air. In the combustion of oil we are able to supply more nearly that amount of air which is actually chemically required.

24 From the manner in which the oil is burned it is evident that a more perfect mixture of air with the gases is obtained. Not only is the air supply easily regulated by hand, but apparatus has been devised to effect the control mechanically and automatically to suit the momentary load requirements. Although it does not necessarily follow that a smokeless chimney signifies complete or most economical combustion, it does mean much to our neighbors and one can safely say that such a chimney is more often obtained in connection with an oil-burning furnace than with one burning coal.

25 Evaporative tests have shown that for good coal an equivalent evaporation of 11 lb. represents about the highest average, although there are cases of higher results on record. For oil the highest average yet attained is probably an equivalent evaporation of 16 lb.

26 In the report of the Naval Liquid Fuel Board the best evaporation with coal is given as 10.2 lb., while with oil the best is 14.4 lb. In some forcing tests made some time ago by F. W. Dean,[1] which are remarkable for the good results obtained at that time, an evaporation for coal of 11.32 lb. was had. In a water-tube boiler under ordinary conditions, using good coal, an average equivalent evaporation of 10 lb. can be considered good practice, although 11 lb. and slightly over is often obtained. Such results as these, it must be borne in mind,

[1] F. W. Dean. The Forcing Capacity of Fire Tube Boilers, Trans. Am. Soc. M. E., vol. 26, p. 92.

can be obtained only with the best grades of steam coal, similar or superior to the grade selected by the writer for comparison.

27 With oil fuel, in connection with water-tube boilers, records show an equivalent evaporation of 14 to 16 lb. Test records in the writer's possession range from 13.6 to 15.4 lb. The latter is the record of the performance of a 150-h.p. Heine boiler in San Francisco. The oil used had a calorific value of 18,629 B.t.u., gage pressure 119 lb., feed temperature 69 deg., flue gas temperature 480 deg., percentage of steam used for burner 2.8 of total water evaporated or 0.36 lb. of steam per lb. of oil; 15.4 is the value after correction for steam to burner. Incidentally the efficiency of the unit is 79.8 per cent. The figures given in the foregoing for oil can hardly be said to be due to any extraordinary construction or arrangement of furnace although care may have been shown in the selection of the burner.

28 In a series of eight tests made with two 66 in. by 16 ft. horizontal return tubular boilers, having one hundred and seventy-six 2-in. tubes each, and using Coalinga oil of 22 gravity, 18,900 B.t.u. per lb. of oil, the average evaporation of water from and at 212 deg. was 15.32 lb. The average total horsepower developed at rate of 34.5 lb. of water per hour was 262.5 for the battery. The theoretical evaporation of the Coalinga 22 gravity oil, 18,900 B.t.u. per lb., being 19.56 lb. of water per lb. of oil and the actual results having been an average evaporation of 15.32 lb., the fuel efficiency for this particular battery was 78.5 per cent.

29 It is the writer's opinion that with a few more years' experience and a more refined and perfected fireroom and furnace arrangement, we will be able to reach evaporations closely approximating the theoretical limits. The ease of manipulation of the fire and the possibility of practically exact regulation of air supply for complete combustion, lend to the fuel oil evaporative qualities that can be found in no other raw fuels.

THE RELATIVE VALUE OF LIGHT OIL AS COMPARED WITH FUEL OIL

By Joseph Nisbet LeConte, Berkeley, Cal.

Member of the Society

30 Crude petroleum consists principally of various combinations of hydrogen and carbon together with comparatively small amounts of nitrogen, oxygen and sulphur. The nitrogen and oxygen and any

incombustible residue or ash may be classed as inert impurities. The sulphur, though combustible, has a low grade of heating value and is otherwise injurious.

31 Taking hydrogen and carbon as the principal constituents, it is found that those oils which are rich in the former element are of light specific gravity as compared with those rich in carbon. The range in specific gravity of California oils may be taken as from unity to 0.84, or from 10 deg. to 36 deg. of the Baumé scale. The majority of the fuel oils will range from unity to 0.9, or from 10 deg. to 23 deg. Baumé. It is also evident that, other things being equal, oils rich in hydrogen will contain more heat units per pound than those rich in carbon. Pure hydrogen contains 62,000 B.t.u. per lb., as compared with 14,500 B.t.u. per lb. for carbon. If petroleum were composed wholly of these two elements, a very consistent law might be expressed between the heat units per pound, and the specific gravity. As a matter of fact the other substances occurring in varying amounts destroy any exact relation.

32 Water in emulsion in crude oil not only acts as an inert impurity in a sample under test, but must be converted into steam in the furnace and thus still further reduces the heat value of the fuel per pound. It occurs in such variable and often in such large amounts, that no relation whatever between specific gravity and B.t.u. per pound can be discerned unless it is eliminated. In the following tables and diagrams, therefore, the oil is assumed to be anhydrous, the water having either been removed before testing, or else its amount determined and corrections made.

33 Nitrogen and oxygen are also inert impurities, but since their amounts are small, their effects are averaged in with the plotted results. Sulphur is a substance which causes considerable variation in the heating value of fuel oil. Sulphur contains 3900 B.t.u. per lb. Oils rich in sulphur, therefore, have a lower heating value per pound, other things being equal.

34 Many determinations of heating values and specific gravities of California oils have been made, but those upon which Table 3 is based were made in the chemical laboratories of the University of California and furnished through the kindness of Prof. Edmund O'Neill and his assistants. The calorific values were determined by means of an Atwater bomb calorimeter, and the specific gravity by means of a Westphal balance or a pycnometer flask at about 63 deg. fahr. Water was determined by distillation.

35 The resulting values for the heavier fuel oils the writer has

collected and plotted, as shown in Fig. 1. The ordinates in this case are degrees of the Baumé scale, which is related to the true specific gravity by the relation

$$\text{Specific Gravity} = \frac{140}{130 + B}$$

or

$$B = \frac{140}{\text{specific gravity}} - 130$$

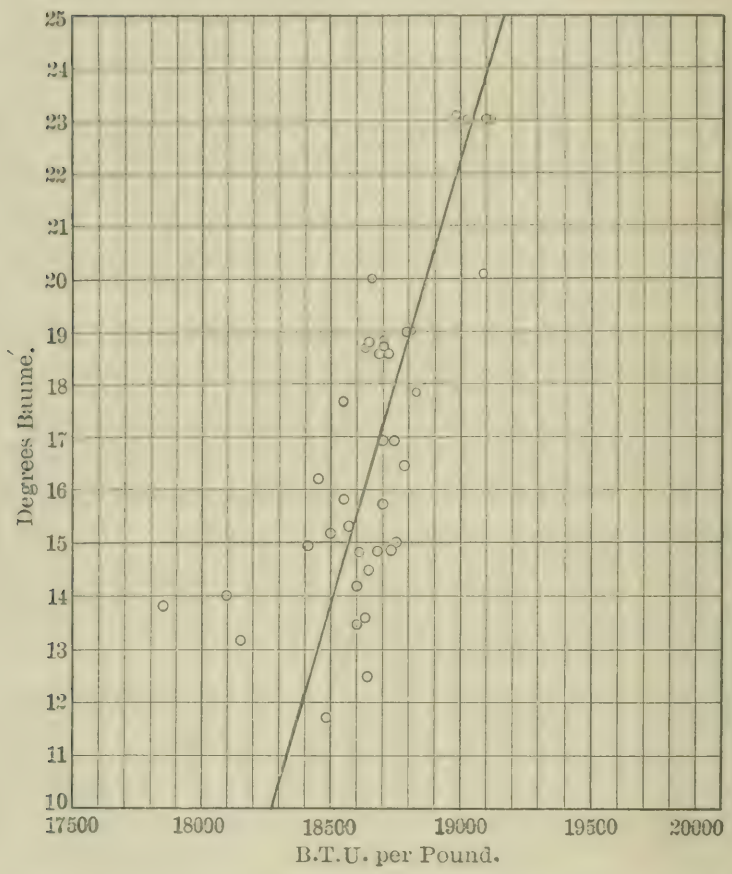


FIG. 1 RELATION BETWEEN SPECIFIC GRAVITY AND B.T.U. FOR HEAVIER FUEL OILS (ANHYDROUS)

at 63 deg. fahr. The abscissae are British thermal units per pound of the anhydrous oil. The points, as might be expected, are rather scattered, but can be approximated roughly by a straight line. In drawing the average line through these points the writer has been guided for its position wholly by the points themselves, and for its inclination, by the points and by a table of average specific gravities and calorific values furnished by R. W. Fenn of the Union Oil Com-

pany, covering a wider range of specific gravities than do the determinations gathered by the writer. The average line gives the relation

$$\text{B.t.u. per lb.} = 17{,}680 + 60B$$

over the range of these measurements.

TABLE 3 SPECIFIC GRAVITY, WEIGHT AND HEAT VALUE OF OIL
(UNION OIL COMPANY)

Deg. Baumé	Specific Gravity	Weight per Bbl.	B.t.u. per Lb.	B.t.u. per Bbl.	Deg. Baumé
10	1.0000	350.035	18,280	6,398,600	10
11	0.9929	347.55	18,340	6,374,100	11
12	0.9859	345.10	18,400	6,349,800	12
13	0.9790	342.68	18,460	6,325,900	13
14	0.9722	340.30	18,520	6,302,400	14
15	0.9655	337.96	18,580	6,279,300	15
16	0.9589	335.65	18,640	6,256,500	16
17	0.9524	333.37	18,700	6,234,000	17
18	0.9459	331.10	18,760	6,211,400	18
19	0.9396	328.89	18,820	6,189,700	19
20	0.9333	326.69	18,880	6,167,900	20
21	0.9272	324.55	18,940	6,147,000	21
22	0.9211	322.42	19,000	6,126,000	22
23	0.9150	320.28	19,060	6,104,500	23
24	0.9091	318.22	19,120	6,084,400	24
25	0.9032	316.15	19,180	6,063,800	25

36 Table 3 gives the computed values of B.t.u. per pound, and specific gravity, as well as the weight per barrel, and B.t.u. per barrel, as determined from this average line. From these it is seen that although the average heating value per pound of crude oil increases as the specific gravity diminishes, it does not increase so rapidly as the weight per unit of volume diminishes. The heating value per barrel of the heavier oils is therefore greater than that of the light ones.

FURNACE ARRANGEMENT FOR FUEL OIL

By C. R. Weymouth, San Francisco, Cal.
Member of the Society

37 In a paper[1] presented at the New York meeting of this Society, December 1908, the writer gave various data regarding California

[1] Trans. Am. Soc. M. E., vol. 30, p. 797.

fuel oil, air required for combustion, the various losses in oil firing, and a description of an automatic system for the regulation of boilers fired with crude oil. Although the present subject properly includes the question of air supply for combustion, the writer, by reason of his former paper, will devote himself mainly to the question of furnace design, and this only for stationary water-tube boilers.

38 About twelve years ago the low cost and certainty of supply of crude oil led to its general use in California boiler plants. At that time with a few notable exceptions, there were none but the crudest methods for burning oil, the owner usually employing one of the numerous improvisors of oil burners to convert his plant, the operation generally consisting in introducing a burner through the fire doors, and covering the grates with fire brick. This plan was occasionally modified by the introduction of different forms of fire-brick arches, target walls, checker walls, etc. Many of these converted coal-burning furnaces caused frequent burning out of boiler tubes, due to the localization of heat; gave a very limited overload capacity; and by reason of various defects of furnace design, coupled with a general ignorance of the question of oil burning, produced in most instances mediocre results.

39 During this early period, E. H. Peabody, then testing engineer of the Babcock & Wilcox Company, began the first extensive engineering investigation of the merits of various types of furnaces, burners, etc., and after an extensive series of tests, lasting nearly two years, developed an oil furnace bearing his name, now in general use with certain types of water-tube boilers.

40 Mr. Peabody's tests indicated the following conclusions:

a That while there are differences in various types of burners these are relatively unimportant.

b The proper design of furnace is of supreme importance as determining efficiency and capacity of boilers and immunity from shutdown, owing to tube burn-outs, etc.

c Fire-brick arches and target walls are not only needless in securing high furnace efficiency, but a menace to continuity of operation, owing to the impossibility of even the best grades of fire brick withstanding the intense heat of an oil furnace.

d Furnace depth and furnace volume are determining factors affecting furnace efficiency and capacity, and affording protection to the boiler heating surface, particularly

when the character of boiler feed water gives rise to an accumulation of scale inside boiler tubes.

e The shape of furnace and path of flame must be such as to provide a nearly uniform distribution of furnace heat over the largest possible portion of boiler heating surface, as distinguished from an arrangement causing the direct impingement of flame on a few inches of the tube length,

f A large surface of heated fire brick is essential to the maintenance of a high furnace temperature and complete combustion of oil.

g The air for combustion should be admitted through carefully planned openings in the floor of furnace in such manner as to provide the most intimate contact of oil flame and incoming air, and thus reduce the air supply to a minimum.

h The flat flame or fishtail burner provides for the most economical use of air for combustion.

i When all these requirements for highest economy are observed, the furnace flame has not the bright incandescence sought by the pioneers in oil burning, but borders surprisingly on an orange red.

41 In the Peabody furnace the bridge wall is set back from the boiler front to give a depth of from 8 to 10 ft., depending on the size of boilers. The burner is of the back shot type, inserted from the boiler front under the floor of the furnace and turning up at the bridge wall. It shoots the flame forward toward the front of the boiler, where there should be an extra course of fire brick set in place without fire clay, to afford added protection to the front wall. With boilers having tubes inclined downwards from the front towards the rear, there is thus provided a furnace design of such a shape as to give the necessary increased volume as the velocity of the flame decreases in flowing away from the burner. This provides a gradual distribution of the furnace heat over the tubes the full length of furnace, without a direct impingement of flame at any point.

42 Except in special boilers, the furnace should have a height at its front end of not less than 6 ft. and for large size boilers the height should be from 7 to 9 ft., depending on the character of feedwater and desired overload.

43 Under ordinary firing, the flame should not extend into the tubes, but under forced firing it will extend part way through the first pass.

44 Under Babcock & Wilcox boilers, Mr. Peabody's record performance of 1903 was 83 per cent efficiency at rating, based on 10 sq. ft. of heating surface per boiler horsepower, and an overload capacity of 110 per cent above rating.

45 When admitting a large excess of air and an ordinary amount of oil, the flame length will be a minimum, and the temperature of incandescence will be reached at the surface of the envelope separating the vaporized oil and air for combustion. This bright flame is sought by the untrained fireman, but it results in a large loss of fuel, as the subsequent mixture of the products of combustion with the excess of air not in contact with the flame produces a lower mean furnace temperature. With economical firing the flame lengthens before coming in contact with sufficient air for complete combustion, and with the highest furnace efficiency this temperature varies from 2500 to 2800 deg. fahr.

46 The location of the furnace relative to the boiler heat absorbing surface is of the utmost importance, not only on account of the loss of heat and consequent radiation from furnace walls when there is excessive travel, but also by reason of the large amount of heat absorbed by direct radiation as distinguished from convection. The first pass of the boiler should be located directly over the furnace, providing the most direct transmission of the heat generated, both by convection and absorption of radiant heat.

47 Owing to the large area of incandescent fire-brick surface, the radiant heat is uniformly diffused over a large heating surface and the amount of heat thus absorbed becomes an important factor in determining the efficiency of boiler heating surface; for while no heat can ultimately be lost, the greater the heat absorbed in the first pass of boiler, the lower will be the temperature of gases on entering the second pass; and finally the later passes of the boiler are able to accomplish greater cooling of the products of combustion, resulting in the lowest possible stack temperature and hence the maximum absorption efficiency of boiler.

ATOMIZATION OF OIL

By A. M. Hunt, San Francisco, Cal.
Member of the Society

48 In order that petroleum may be burned with complete combustion, it is necessary that it be either gasified or injected in the form of a spray into the furnace in which it is burned. If the oil is being injected into a coal furnace, that is, one not enclosed by walls of brick or other material which becomes highly heated, the oil must be injected in a spray composed of very fine particles, in order that none of the particles shall fall to the bottom unconsumed. If the walls are highly heated, the radiation from them will aid greatly in vaporizing the oil particles, and larger particles will be consumed before they drop. If the furnace is short, the oil particles will have a relatively short time period within which they must be consumed, and must, therefore, be smaller.

49 It is evident that anything which will enable the oil more easily to be atomized, or that will aid in vaporizing an oil particle after it is injected into the furnace, will help to reduce the quantity of atomizing medium required, whether it be steam or air.

50 Most of the oils used for fuel are of a heavy and viscous character, and their viscosity is rapidly reduced by rise in temperature. It is, therefore, desirable that the oil fed to the burners be preheated, and it is almost universal practice to do so. The preheating is usually accomplished by passing the oil through a heater similar in type to a closed feed-water heater, using the exhaust steam from the pumps handling the oil. The burner should be so built that the relative areas of openings for issue of oil and atomizing medium can be maintained, or if they become enlarged by scoring, that adjustment can be readily and inexpensively made.

51 Some work has been done in the direction of atomizing the oil without the use of air or steam. The oil having been preheated to temperatures of from 220 to 260 deg. fahr., is injected into the furnace through a needle nozzle, having a small sized orifice. The portion of the needle stem inside the cylindrical part of the nozzle has cut on it a screw thread, which imparts to the issuing oil a rotary motion. The release of pressure on the heated oil, and the rotary motion imparted to it, cause it to issue in the form of a spray in a state of subdivision fine enough to enable it to burn successfully.

52 I give below some data as to amounts of atomizing medium required for fuel oil, obtained from various sources:

DATA AS TO AMOUNT OF STEAM REQUIRED FOR ATOMIZING FUEL OIL IN BOILER FURNACES

CASE 1	
Steam supplied to burner per lb. of oil	0.537
Actual evaporation per lb. of oil	13.48
Percentage	4.0
CASE 2	
Steam supplied to burners per hour	4373.0
Actual evaporation per hour	96881.0
Percentage	4.5
Steam per lb. of oil for atomization	0.529
CASE 3	
Steam supplied to burners per hour	7087.0
Actual evaporation per hour	174820.0
Percentage	4.0
Steam per lb. of oil for atomization	0.485
CASE 4	
Steam supplied to burners per hour	5746.0
Actual evaporation per hour	144079.0
Percentage	4.0
Steam per lb. of oil for atomization	0.475
CASE 5	
Steam used was measured by use of separate boiler.	
Oil used	$34\frac{1}{2}$ deg. Baumé
Total evaporation per lb. of oil from and at 212 deg	14.99
Percentage steam evaporated, used by burner	7.4
CASE 6	
Steam used was measured by calibrated nozzle.	
Total evaporation per lb. of oil from and at 212 deg	14.7
Percentage steam evaporated, used by burner	2.5
CASE 7	
Steam used was measured by calibrated nozzle.	
Total evaporation per lb. of oil from and at 212 deg	14.2
Percentage steam evaporated, used by burner	3.6
CASE 8	
Steam used was measured by use of separate boiler.	
Total evaporation per lb. of oil from and at 212 deg	15.2
Percentage steam evaporated, used by burner	2.96

53 From tests made under the direction of the Bureau of Steam Engineering in 1902, the following data are taken: Four tests were made using steam as the atomizing medium. The percentage of

total evaporation used by the burners ranged from 3.98 to 5.77 per cent. A number of tests made under Stirling water-tube boilers gave results ranging from 2.1 to 3.42 per cent.

54 From the above data and general practice and experience, the following statement can be made: In designing a plant it is entirely safe to assume 5 per cent of the evaporation of the boilers for steam supply for burners. In operation, if the amount is greater than 3 per cent, it may be concluded that the condition can be bettered.

55 The use of compressed air for atomizing fuel oil may be stated to offer no opportunity for fuel saving over the use of steam direct in cases where steam is available. The direct use of steam obviates complication, and risk of interrupted service, and the use of compressed air is not justified unless there is some special reason for it. It may be that steam cannot readily be had, and that a motor-driven compressor can easily be installed, or where steam is available, the loss of water through steam supplied to the burners may be undesirable, as in the case of sea-going steamers.

56 In certain metallurgical and industrial operations, especially where high temperatures are desirable, the use of air is to be preferred. In rotary cement kilns, using oil fuel, compressed air as the atomizing agent is universally employed so far as I know. Reverberatory furnaces for metallurgical operations, using oil fuel, employ compressed air.

DATA AS TO AMOUNT OF AIR REQUIRED FOR ATOMIZING FUEL OIL

Case 1

Rotary cement kiln, 7 ft. 6 in. in diameter by 125 ft. in length, producing about 500 bbl. of clinker daily. Air used under pressure of approximately 80 lb. Single burner, delivering oil at the rate of about 4 gal. per min. The weight of air required for atomization is approximately 25 per cent of the weight of the oil atomized.

Case 2

Data furnished by air compressor manufacturer as to capacity of compressor furnished for boiler installation of 2500 h.p. Assuming that the entire capacity of the compressor was necessary, this figures out a use of air amounting to approximately 55 per cent of the weight of fuel atomized.

The same manufacturer gives the following data: For marine boilers figure 1 cu. ft. of free air for each 5 boiler h.p., the air being supplied at a pressure of about 25 lb.

A 10-cu. ft. compressor at 20 lb. will supply air sufficient to atomize the fuel oil to take care of a boiler furnishing steam heat for a small apartment house.

For burning sewer pipe it takes about 5 cu. ft. per min. for each burner using about 5 gal. per hr. of oil.

He adds the following comment: We favor the use of air as hot as we can get it from the compressor, cutting out the water jackets.

CASE 3

Reverberatory copper matting furnace. Hearth 80 ft. long by 17 ft. wide. Center of roof arch is about 39 in. above the surface of bath on hearth. Fired from the end using four burners. Amount of material smelted per day, 182 tons. Oil used per day, 36,472 lb. Air for atomizing supplied by a motor-driven Connersville blower at 9 lb. pressure. Amount of air used, about 50 per cent of weight of oil burned.

SIZE OF STACKS WITH FUEL OIL

BY K. G. DUNN, SAN FRANCISCO, CAL.

Member of the Society

57 The question of stack area and draft depends on the quantity of fuel burned and the draft required. In coal burning, the draft necessary to overcome the friction of the fuel bed runs anywhere from 35 to 70 per cent of the total draft head. This is done away with in oil-burning furnaces and consequently a shorter stack will answer. A height of stack of from 80 to 100 ft. is all that is necessary. In coal-burning plants stack sizes are based on 5 lb. of fuel per boiler h.p. In changing to oil we have to figure on the basis of only $2\frac{1}{2}$ lb. per boiler h.p., and considering that oil can be burned with a smaller amount of excess air than coal, a stack of given area would serve for double the horsepower with fuel oil than it would with coal.

58 There are many stacks operating successfully on the basis of 50 per cent of the sizes given in Kent's table. I have in mind a record of one stack which, when forced to the limit, operated on the basis of 35.7 per cent. When this point is reached the pressure in the furnace is so great that the gases pass out through the joints in the setting and through the boiler breeching. One point in connection with this, which I believe is not given the attention that it should have, is that of breeching area. Ordinarily the effective area of the breeching is not considered, as in many plants the breechings are narrow and long, and in several instances that I know of, the effective areas are in the neighborhood of 70 to 80 per cent of what they really should be.

LOCOMOTIVE PRACTICE IN THE USE OF FUEL OILS

By Howard Stillman, San Francisco, Cal.

Member of the Society

59 About the first practical use to which fuel oil was applied as a means of generating steam in California was on a locomotive engine of the Central Pacific Railroad, in 1886, the tests being made between Sacramento and Davisville. The oil was comparatively thin and of a reddish color, quite different from the heavy asphalt oil now in use. Its origin I am unable to locate at this time. The experiments were quite satisfactory and demonstrated that a working steam pressure could be maintained in a locomotive boiler in freight service with oil fuel. The cost of this oil was high and the matter was dropped. This was previous to the development of petroleum products in California.

60 Following the subsequent oil developments in California and the promise of a sufficient supply, a continuation of the early experiment was made and tests indicated great possibilities for oil fuel. The first regular conversion of a locomotive to oil burning in regular service was on the Southern Pacific Railroad in November 1900 and a number of tests made in comparison with coal fuel proved so satisfactory that, in February 1901, we were authorized to equip other engines. This was gradually done, and in about five years all locomotives were converted to oil burners on the Southern Pacific in California, and extending to El Paso, Texas, embracing what is known as the Pacific System Lines. The approximate number of locomotives we now have burning oil is as follows:

Passenger	443
Freight	533
Switch	177
Total	1153

61 On the above basis the consumption is 788,687 lb. per month, or 9,464,244 bbl. per year.

62 The characteristics of the oil have been fully covered in the previous papers. There is, of course, a range in variation in specific gravity of 14 deg. or 15 deg. Baumé. Its calorific value

is taken at 18,500 B.t.u., which we consider to express its mean heat value. The following specifications for fuel oil have been adopted by the Southern Pacific system:

General Requirements. Liquid fuel is crude petroleum as received from the wells, or the product of crude petroleum, distilled or reduced. It must contain no sand or foreign matter in the shape of sticks, waste, stones, etc., and must be sufficiently liquid to flow readily in 4-in. pipes at a temperature of 70 deg. fahr. It must contain as little water as possible, and oil containing more than 2 per cent of water and other impurities will not be accepted.

Fuel oil will be paid for on the basis of volume at 60 deg. fahr., also deducting all water contained, according to method outlined as follows:

Tests. One sample will be taken from each carload or fraction thereof. The sampling of cars is to be made with car thief having valve at lower end. The thief with open valve will be lowered gradually into car and valve closed at instant of touching bottom. The thief thus filled will contain oil sample to be tested for water, sand, basic sludge, and specific gravity.

Oil received in settling or storage tanks will be sampled with Robinson or other standard thief, a sufficient number of samples being taken to secure an average of its contents.

Fuel oil will not be accepted for general use, the flash point of which is less than 110 deg. fahr. when tested by the open cup, Tagliabue method. The oil to be heated at rate of 5 deg. per min., and test flame applied every 5 deg., beginning at 90. This flashpoint being the danger point at which the oil begins to give off inflammable gas, the fire or burning point is not required.

The test for water, sand, and Baumé specific will be made as follows: 100 cu. cm. of the sample will be placed in a 250-cu. cm. graduated glass cylinder provided with stopper, and thoroughly shaken up with not less than 150 cu. cm. of gasoline. The mixture will be heated to 120 deg. fahr. for from 3 to 6 hours to facilitate the separation of impurities, the amount of which can then be read from the graduations of cylinder. All proportion of water and other impurities contained in the sample will be deducted from the volume contained in the car and not paid for.

The temperature of shipment will be tested directly as sample is removed from sampling tube, or by immersion of thermometer in the receptacle itself for not less than one minute. A deduction in volume for expansion at temperature of over 60 deg. fahr. will be made at rate of 0.0004 for each degree. At 90 deg., the deduction would be $1\frac{1}{5}$ per cent, etc.; Kansas and Oklahoma fuel oil furnished from Sugar Creek or Kansas City, Mo., at 90 deg., should have a deduction of $1\frac{1}{4}$ per cent.

Gravity of fuel oil should range between 13 and 29 deg. Baumé at 60 deg. fahr.

Conditions. If any portion of an accepted shipment is subsequently found to be damaged, or otherwise inferior to the original sample, that portion will be returned to the shipper at his expense. Any sample failing to meet all the requirements of this specification will be condemned, and the shipment represented by it will be returned to the manufacturers, they paying freight both ways.

63 In the evolution of oil burning in locomotives, the matter unfortunately resolves itself to one of local conditions. The modern locomotive boiler is not an ideal form of oil furnace. Railroads must burn whatever fuel is cheapest and most available, and the matter continues to be one largely of expediency.

64 Locomotive boilers are designed for coal fuel and in the early days of converting engines to liquid fuel we claimed that should emergency arise the oil burning engines could be converted over night in the roundhouse back to coal again. From the figures given us by Mr. Bell as to extent of petroleum deposits in this state, it does not seem at all likely that the contingency will soon arise.

65 Locomotive fuel oil is carried in tanks built to fit the coal space in the tender. Additional flat tanks when required are placed over the coal space or back of it. Gravity supply is depended upon through flexible pipes to the locomotive. By use of steam coils the oil in tender tanks is heated to about 120 deg. Oils with less viscosity than California fuel oil would require less preheating. Each system of oil tanks on the tender is provided with a gage board or scale from which the fuel records are kept.

66 The burner used is of the flat jet non-pressure type consisting of a flat casting divided longitudinally by a partition over which the oil flows as it is admitted to the upper cavity. The lower cavity receives the steam for the jet which strikes the oil flowing over the partition, spraying it into the furnace. We aim completely to atomize the oil near the burner tip in order that it may be immediately vaporized. The form of burner is of minor importance provided it is simple, easy to clean, and without complication from carbonizing. It has in truth been said that those who try to improve the efficiency of fuel oil by alteration of the burner are on a plane with those who try to improve the steaming quality of a boiler by altering the injector. High efficiency from fuel oil is due mainly to the arrangement of the furnace.

67 The steam for atomizing is obtained from the dome and is available at full boiler pressure of 200 lb. through a suitable regulating valve. We have used compressed air experimentally and for some time used a form of burner that delivered air inductively to the burner itself. Other than by a localization of heat at the point of the burner, no benefit could be found by tests with air mingled with the steam in this way. Atomization with compressed air is undoubtedly of value under certain conditions but is liable to produce locally in the furnace a more intense heat than is desirable. With a dry or

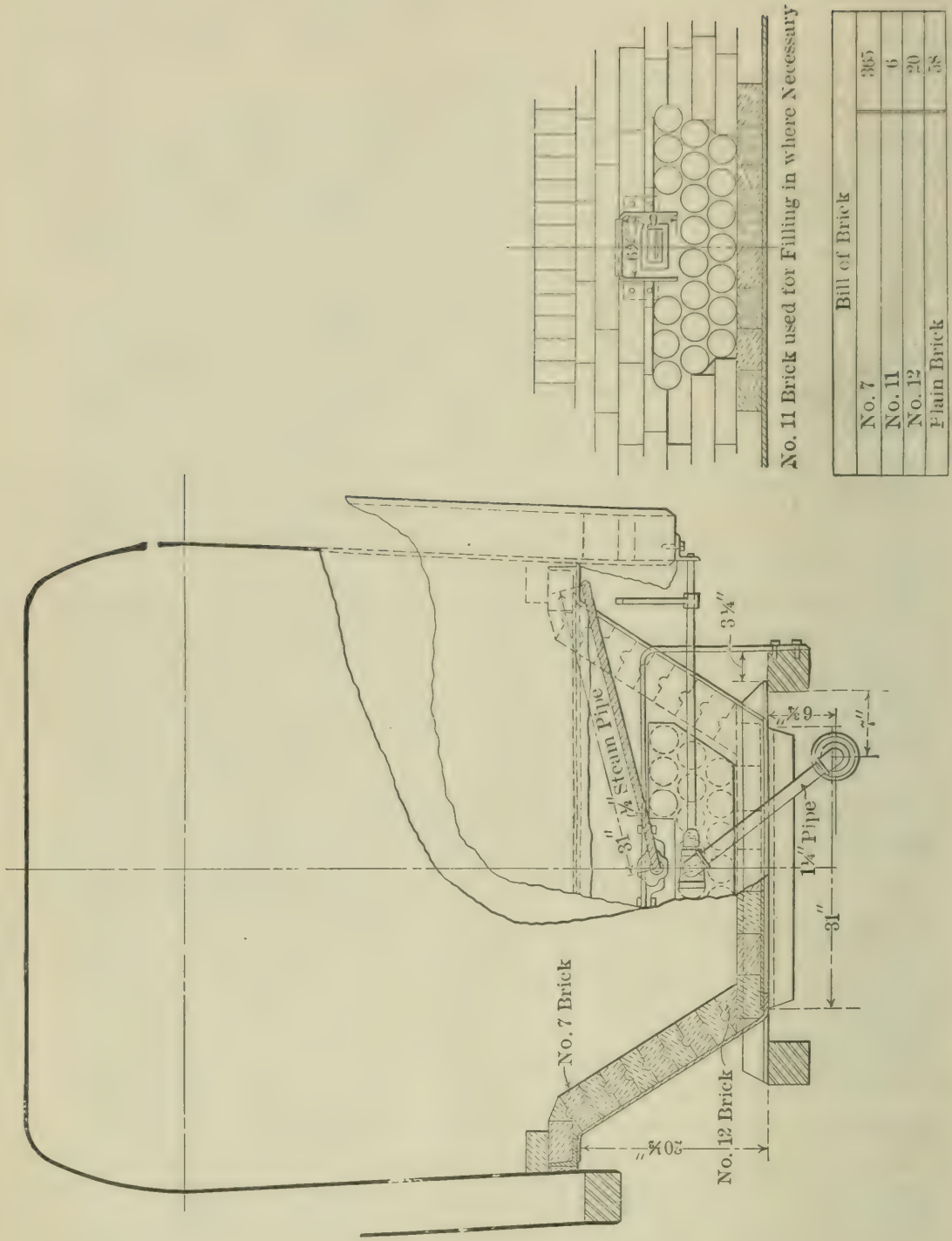


FIG. 2 CROSS-SECTION OF LOCOMOTIVE FIREBOX, SHOWING METHOD OF FITTING FOR OIL BURNING

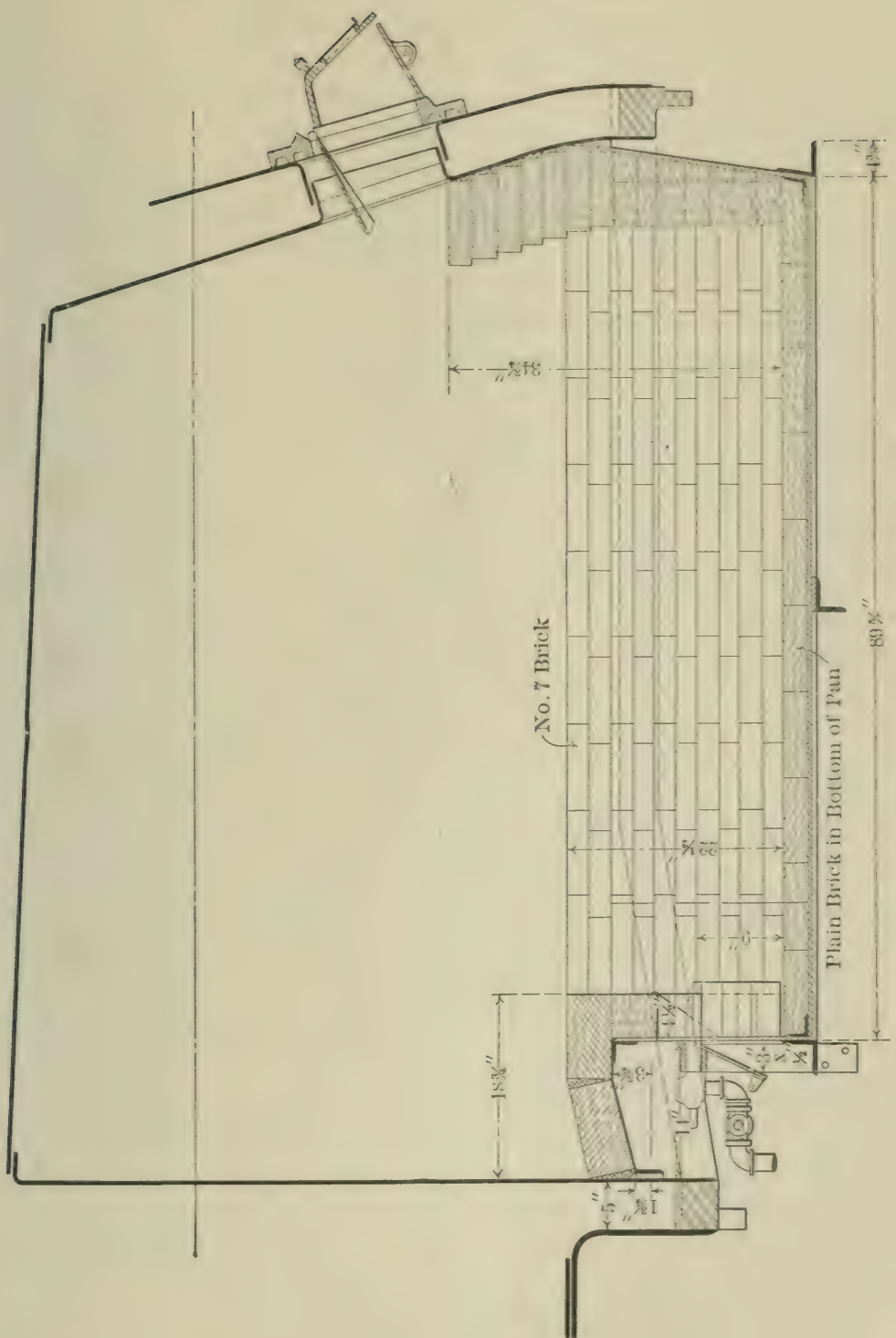


FIG. 3 LONGITUDINAL SECTION OF FIRE BOX

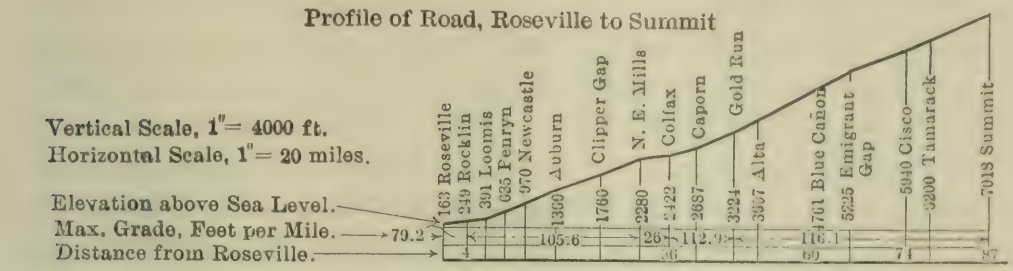
superheated steam jet the oil is sprayed and broken up so as to allow the air admitted through proper dampers to mix and the oil to be consumed completely without damage to the sheets. Tests on our locomotives by Professor Grey, formerly of the University of California, show that temperatures ranging from 2500 to 2750 deg. fahr. are obtained, the latter being the highest observed.

68 The method of fitting a locomotive fire box for oil burning is shown in Figs. 2 and 3, which is the present standard arrangement for our heavy freight engine of the consolidated type. Fire bricks, the most refractory obtainable, are placed at the lower side of the fire-box plates to prevent impinging of the oil blast against the sheets. No more bricks than are necessary for the purpose are used. The most refractory bricks melt out in a comparatively short time—not from an intense degree of heat so much as from the fluxing agents introduced with the oil, especially salt or other alkalis with which California petroleums are associated.

69 Fire-box repairs to oil-burning locomotives do not exceed those of coal-burning engines. From records kept of cost of maintenance, we know that if a furnace is properly equipped with draft adjustment and burner, an oil-burning fire box should outlast a fire box burning coal; that is, on the same class of engines in the same service. However, it cost a good many fire boxes to determine the best arrangement. In this connection I wish to emphasize the fact that the oil fireman is a large factor in successful oil burning on locomotives. The condition is very different from that of oil firing on stationary boilers. Conditions are constantly varying in the locomotive engine, depending on load and speed, regularity of service, condition of track, grade and rules of the road. The oil fireman must necessarily watch the operation of the locomotive with intelligence and every movement of the engineer demands corresponding regulation of the fire and watchfulness on the part of the fireman. He has two gages to guide him, the steam gage and top of the stack to show results, the desired steam pressure with least smoke being the objective. It is a general though not universal condition that the locomotive should give some smoke. In general a thin grey smoke is indicative of the most economic result. A careless fireman can do great damage to the fire box and flues if he neglects to attend to the dampers and regulators at the proper time.

70 It is my opinion that a greater depth of fire box is more essential for economic oil burning than for coal. If combustion is not approaching completion when the gases enter the flues, the vapors in

TABLE 4 DATA FROM EVAPORATIVE TESTS OF LOCOMOTIVES CROSSING THE SIERRA NEVADAS, SACRAMENTO DIVISION, SOUTHERN PACIFIC COMPANY



Class of engine	10 wheel	Consolidation	Mallet consolidation
Service	Passenger	Freight	Freight
Date of test	May 21 and 26, 1908	June 9 and 11, 1909	Nov. 6 and 8, 1909
Number of single trips in test	2	2	2
Total time of test	17 hr. 39 min.	21 hr. 23 min.	29 hr. 2 min.
Actual running time	13 hr. 55 min.	12 hr. 59 min.	18 hr. 11 min.
Miles run	315	174	174
Average steam pressure (gage), lb	196.0	196.1	194.5
Smoke box temperature, deg. fahr.	797	738.5	451.3
Total gal. water evaporated	44147	48103	91087
Total lb. water evaporated	367642	400858	759058
Total gal. oil burned	3951.6	4328.4	7692.1
Total lb. oil burned	31613	34627	61537
Equivalent evaporation, lb. water per lb. oil	14.14	13.95	15.04
Lb. water evaporated per sq. ft. heating surface per hr.	8.698	8.809	6.392
Lb. oil burned per sq. ft. heating surface per hr.	0.748	0.761	0.518
Number of cars in train	7	14.5	24.5
Weight of train, tons	342	481	1056
Gross ton mileage	107730	83694	183744
Gal. water evaporated per 1000 ton mi.	409.79	574.75	495.73
Lb. water evaporated per 1000 ton mi.	3413	4790	4131
Gal. fuel oil burned per 1000 ton miles.	34.90	48.40	39.51
Lb. fuel oil burned per 1000 ton miles.	279.20	387.20	316.08
Boiler efficiency, per cent	73.84	72.83	78.52
Maximum i.h.p.	1719	1470	2486
Mean i.h.p.	1368	1222	2057
Dimensions of locomotive			
Engine no.	2315	2564	400
Size of cylinders, in.	22x28	22x30	26 and 40x30
Diameter of drivers, in.	63	57	57
Total weight of locomotive, lb.	203300	208000	425900
Weight on drivers, lb.	160000	187000	394150
Weight of tender, lb.	138070	134745	169765
Total heating surface, sq. ft.	2994	3403	6394
Feed water heater, heating surface, sq. ft.			1221

NOTE.—Train weights are the average for the distance hauled and are exclusive of engine and tender. Tests made under ordinary service conditions. Engines unaided. In "water evaporated and oil burned per sq. ft. heating surface per hr.," the figures are for actual running time; the allowance of 15 gal. of oil and its equivalent in water is made per hr. while standing. In "gal. and lb. oil per 1000 ton mile," a deduction is made of 15 gal. per hr. while standing and 3.5 per cent of oil for evaporating steam for atomizing oil. Quantity of oil burned corrected to normal temperature of 70 deg. fahr. All measuring instruments calibrated. Analysis of fuel: Kern River oil; gravity, 15.8 deg. Baumé; flash point, 230 deg. fahr.; fire point, 278 deg. fahr.; commercial weight, 8 lb. per gal.

process of combustion fall in temperature below that required for oxidation of the carbon, and it is precipitated as soot or black smoke. While this is a common phenomenon of smoke production with any fuel, it is more in evidence with oil than with solid fuel where there is nothing to burn but volatile combustible. Lack of oxygen is generally supposed to be the cause of smoke, but lack of sufficient temperature required for chemical combination is as often, if not oftener, the cause. The lack of sufficient fire-box volume is, in my opinion, the common cause of smoke in oil-burning locomotives when the greater demands are made on the boiler for rapid evaporation. The deposit of soot upon locomotive flues in oil burning is a common difficulty and calls for the regular operation in service of sanding the flues. Soot is a poor conductor of heat and the steaming quality of a locomotive rapidly falls off when flues become lined with soot. The operation of sanding consists in dropping the reverse lever as much as possible, while the engine is running, opening the throttle wide, while the fireman puts a few cupfuls of sand through the opening in the fire door, the draft carrying the sand through the flues and ridding them of soot, the operation being repeated if necessary. A supply of sand is carried on the foot board for this purpose.

71 In oil burning the factor of grate surface disappears. The grates are bricked over and air admission regulated through openings and proper dampers. There is no function corresponding to grate surface such as used in connection with coal burning. Only the item of heating surface remains on which to base comparisons.

72 The rate of combustion on the locomotive, like other functions, depends on the service and size of the machine. While the principles of combustion do not vary in locomotive boilers from those in stationary service, there is a wide variation in the rate of steam production, independent of the kind of fuel used. With the modern locomotive in main line service the demand for steam at full pressure per unit of heating surface far exceeds, in point of time, the product of a stationary boiler. The demand on the locomotive is intermittent and conditions vary from an enormous rate of steam production when working at a rate of maximum effort, to one of comparative rest.

73 A large number of locomotive tests have been made by the Southern Pacific Railroad, covering the use of fuel oil separately as well as in comparison with coal. Table 4 shows evaporative results on most recent tests covering about the heaviest oil firing which we have, crossing the Sierra Nevadas.

74 Comparisons of coal and oil from an economic standpoint are

most interesting where the two fuels come in competition, and relative values must largely depend on the conditions and form of boilers under which either fuel is burned. Some engine boilers are better adapted for coal than oil, or vice versa, depending on construction. Comparisons therefore are to some degree a function of the boiler itself. This was gone into quite thoroughly when the locomotives were converted and a summary of the results is given in Table 5. The figures are based on evaporative tests before and after conversion. The comparisons are with ordinary bituminous coal of about 13,350 B.t.u.

75 The total number of locomotives accounted for in Table 5 is 745, and the mean equivalent is 152 gal. or 3.62 bbl. of 42 gal., equivalent in heat value to 2000 lb. of coal. The average

TABLE 5 COMPARATIVE TESTS WITH OIL AND COAL

Type of Locomotive	Number in Service	Evaporation, 2000 Lb. Coal Equivalent to Fuel Oil, Gal.
Eight-wheel 18-24	50	144
Ten-wheel	294	151
Mogul	176	146
Twelve-wheel	67	158
Consolidation	139	162
Atlantic	19	144

equivalent figures in the table cover much careful work and are based on equivalent evaporative results from many service tests. It should be borne in mind that the figures are from service tests and represent oil burned on the main line between terminals only and not losses in transfer, handling or firing up, etc. Our accounting department uses a figure of 168 gal. or 4 bbl. of oil, equivalent to one ton of coal.

76 It may be said that with fuel oil, provided engines are burning it with a fair degree of efficiency, the quality of fuel is a constant factor. Poor or bad coal as accounting for engine failures drops from the delay report. The old excuse, "cleaning fires" does not hold.

77 In addition to engines enumerated in Table 5, we now have 29 Consolidation Freight Mallets and 12 Mogul Passenger Mallets in service. These are all oil burners and have never been used with coal fuel, hence no fuel equivalents can be furnished for engines of this class. Oil is an ideal fuel for Mallet locomotives, as the personal equation is eliminated as relates to physical inability of firemen to maintain coal fires on long runs with engines of this class.

78 Concerning relative values of coal and fuel oil I cannot close without submitting Table 6, showing results with both fuels on the Southern Pacific Company's bay steamers, the figures comprising the last six months' record with coal for the year 1901 compared with the last six months in oil for the year 1908. The figures shown are not from evaporative tests but cover actual service of eleven steamboats, a total of 176,790 miles as coal burners, and 186,771 miles as oil burners, and are from the official accounting record.

TABLE 6 RECORD OF COAL BURNED DURING LAST SIX MONTHS OF 1901, AND OIL BURNED DURING LAST SIX MONTHS OF 1908, ON STEAMERS OF SOUTHERN PACIFIC COMPANY

Steamer	Tons Coal Consumed Last Six Months of Coal Burning	Mileage for Corresponding Period	Miles per Ton of Coal Consumed	Barrels Oil Consumed, Six Months (42 Gal. per Bbl.)	Mileage for Corresponding Period	Miles per Bbl. of Oil Consumed	Equivalent of One Ton of Coal for Equal Mileage in Bbl. of Oil
Berkeley.....	2764	18592	6.72	11207	21130	1.89	3.56
Piedmont.....	4223	19548	4.63	15414	20808	1.35	3.43
Oakland.....	3209	18348	5.72	13603	19894	1.46	3.92
Bay City.....	2889	21536	7.45	11548	20943	1.81	4.12
Encinal.....	3703	20284	5.47	10512	19678	1.87	2.9[illegible]
Newark.....	1860	9372	5.04	6559	10631	1.62	3.11
Transit.....	2581	16715	6.48	9548	17922	1.88	3.45
El Capitan.....	1019	7238	7.10	3431	6072	1.77	4.01
Solano.....	4516	5480	1.21	17151	7143	0.42	2.88
Apache.....	1971	18992	9.64	7996	21380	2.67	3.61
Modoc.....	2265	20685	9.13	7682	21170	2.76	3.31
Totals and Resultant Means	31000	176790	5.70	114651	186771	1.63	

Mean Equivalent of 1 Ton of Coal in Bbl. of Oil, 3435

79 The service of these bay steamers represents a consumption per year of 229,302 bbl., which, added to my former figure per year for locomotives, gives a total of 9,693,546 bbl. per year consumed by the Southern Pacific Company. In addition to this a considerable amount is used in shops and rolling mills.

80 Let us hope that the rate of production in California as shown in Mr. Bell's paper, 63,000,000 bbl. in 1910, may maintain. Approximately one-seventh of the output is consumed by the Southern Pacific Company.

MARINE USE OF FUEL OIL

By J. H. Hopps, San Francisco, Cal.

Member of the Society

81 As a fuel for steamships, petroleum has many advantages besides that of low cost, the most important of these being:

a The saving in labor and consequent reduction in the number of firemen. The amount of money saved varies with the size of the ship and the number of firemen carried. In installations of average size, one-third the number of firemen and coal passers necessary when burning coal would be sufficient.

b Reduction in weight and bulk of fuel, giving increased cargo capacity and resultant greater earning power. Comparing "Wellington Screenings," a type of coal generally used for steamship work on the coast, and fuel oil at from 14 to 17 Baumé, oil for equal heating value occupies about one-half the space taken by the coal and has less than one-half the weight. Oil may be carried in parts of the ship not otherwise useful.

c Saving in time. The time consumed in coaling and expense of moving to bunkers is saved, as fuel oil can be pumped into the ship when at the dock and while the cargo is being taken on or discharged.

d Uniform steaming. The rate of steaming can be kept uniform, there being no loss due to cleaning fires, etc.

e Cleanliness, due to the absence of coal dust and dirt when coaling and to the absence of ashes in the fireroom.

f Reduced cost of maintenance. Fewer repairs on boilers due to uniform temperature in furnace and combustion chamber. No corrosion of floor plates, fire fronts, or bunkers. No grate bars to burn out, fire doors or ash-handling machinery to renew or repair.

82 That the advantages of oil as fuel are recognized is evident from the large number of steamers using this fuel. Some of these vessels are of large size and make long voyages, notably, the ships of the Hawaiian American Steamship Company, the steamers Tenyo Maru and the Chiyo Maru of the Toyo Kisen Kaisha Company, the steamers Sierra and Mariposa of the Oceanic Steamship Company and the numerous large tank steamers owned by the Standard Oil, Associated Oil and Union Oil Companies.

RESULTS OF TESTS

83 It is not possible to give exact figures upon the saving effected by the use of oil as the prices of both coal and oil are constantly changing. Some instances, however, may be quoted.

84 In the report of the Naval Liquid Fuel Board published in 1904, figures are given upon the performance of the Steamship Nevadan of the Hawaiian American Steamship Company. The following tabulation is taken from these, voyage No. 1 with coal being from San Diego to New York and voyage No. 2 with oil from New York to San Diego.

	Total I.H.P.	Fuel	Total Consumption of Fuel	Coal per I.H.P.	Oil per I.H.P.
Voyage No. 1.....	1833	coal	2269 tons	2 lb.	.....
Voyage No. 2.....	2196	oil	9126 bbl.	.....	1.1 lb.

85 Of the coal burned, part was Eureka and part Coronel coal. The heat value of the coal is not given. The figures for voyage No. 2 represent an exceptionally fine performance. The steamer was new, fitted with triple-expansion engines, the Howden system of forced draft, and the Lassoe-Lovekin oil-burning system.

86 When burning oil, six men were required in the fireroom as against fifteen when burning coal. Four hundred and fifty-seven tons of measured space for cargo was saved on account of the decreased bulk of the oil fuel. The financial gain to the company from all causes is given as $500 per day.

87 In the case of a small coasting steamer coming under the observation of the writer, careful records were kept of the fuel cost, both with oil and coal. The cost of fuel per hour of actual steaming, averaged for a period of six months in each case, was

Coal at $5.25 per ton..... $2.65 per hr.
Oil at $0.70 per bbl..... 1.64 per hr.

88 In 1903 a series of tests were made by T. W. Ransom on the tugs Richmond and A. H. Payson, owned by the Santa Fé Railroad Company. As these vessels ply only on San Francisco Bay and in smooth water, the installation of platform scales to weigh the feed water and fuel oil was feasible. The tests were made with great care, a number of observers being employed and all essential data recorded to show the efficiency of the entire installation and of the boilers

and engines separately. The detailed data secured were destroyed in the fire of 1906, there remaining only a summary of the results.

89 The machinery of the two vessels is identical with the exception of the boilers. The Richmond is fitted with a boiler of the Scotch marine type, 13 ft. mean diameter by 11 ft. long, with three Morrison furnaces, 3 ft. 6 in. in diameter by 7 ft. 10 in. long, and 230 tubes $3\frac{1}{2}$ in. in diameter by 7 ft. 10 in. long. The depth of the combustion chamber is 36 in. and the total heating surface is 2136 sq. ft. The A. H. Payson is fitted with a Babcock and Wilcox marine water-

TABLE 7 RESULTS OF FIVE-HOUR TESTS

FIVE-HOUR RUN, TUG A. H. PAYSON, AUG. 2, 1903

Time	R.p.m.	H.p.	Water used	Oil used	Water evap. actual	Water evap. from and at 212 deg.	Factor of evap.	Water per i.h.p.	Oil per i.h.p.	Speed knots
11.00										
12.00	95.3	535	11475	853	13.4	14.68	1.095	21.4	1.59	
1.00	96.0	523	11326	837	13.5	14.77	1.094	21.6	1.60	
2.00	94.5	509	11418	809	14.1	15.32	1.087	22.4	1.58	11.72
3.00	95.5	498	11300	826	13.6	14.66	1.078	22.9	1.65	10.63
4.00	96.4	537	12251	845	14.6	15.84	1.085	22.9	1.56	11.65

FIVE-HOUR RUN, TUG RICHMOND, AUG. 24, 1903

12.00										
1.00	91	418	9532	829	12.2	12.80	1.140	22.8	1.98	
2.00	92	424	10331	835	12.4	14.10	1.143	24.3	1.97	
3.00	91	418	8831	806	11.0	12.55	1.140	21.1	1.92	11.08
4.00	91	418	10496	833	12.2	13.90	1.140	25.1	1.99	10.04
5.00	91	418	9627	865	11.2	12.90	1.150	23.0	2.06	10.75

AVERAGE FOR FIVE HOURS RUN

Payson	95.5	520	11553	834	13.8	15.05	1.089	22.2	1.59	11.15
Richmond	91	419	9743	833	11.8	13.05	1.142	23.2	1.96	10.47

tube boiler, total heating surface 2770 sq. ft. The engines in both cases are compound engines, high-pressure cylinders 20 in. in diameter, low-pressure cylinders 42 in. in diameter, and stroke 24 in.

90 A large number of tests were made on these vessels in actual service when towing car floats to and from Point Richmond. In addition, a 5-hour test running steadily without a tow was made on each boat, with the results given in Table 7.

91 From examinations of the logs of numerous steamships, it appears that with vessels fitted with triple-expansion engines devel-

oping from 1000 h.p. up, with everything in first-class condition, the fuel consumption will be about 1¼ lb. of oil per i.h.p-hr. For smaller vessels fitted with compound engines, the consumption will range from 1.6 to 2 lb. per i.h.p-hr., depending on the efficiency of the plant.

OIL STORAGE

92 In order to render a fuel oil installation safe, careful attention must be paid to the construction of the tanks in which the oil is stored. Not only should the very best workmanship, the best methods of support and the best quality of riveting be insisted on, but great care should be exercised in the design of the ventilation system. Air pipes should be fitted to all tanks of sufficient size to lead off gases as they accumulate and to prevent any undue pressure on the tanks due to too rapid pumping when they are being filled. The ventilating pipes should be led as directly as possible to above the uppermost deck of the vessel. They should not be near the smoke stack and should be placed where it will be impossible for a naked light to be near them. The openings should in all cases be covered with wire gauze carefully secured. Further than this, the workmanship on all pipes, valves and fittings should be of the very best quality. Great care should be taken that all joints are perfectly tight, as the leakage of a very small quantity of oil may result in a formation of a large volume of gas from which a disastrous explosion or a serious fire may result.

93 In the case of wooden vessels, separate steel tanks independent of the structure of the ship are provided. The location of these tanks varies greatly, depending on the trade in which the ship is engaged and the preference of the superintending engineer. They are frequently placed in the space formerly occupied by the coal bunkers, in the fore-peak and, often, on deck.

94 In a steel vessel fitted with double bottom, the fuel oil may be stored in the compartments in the double bottom usually devoted to water ballast, or in deep tanks constructed for the purpose and usually extending entirely across the ship. Where deep tanks are used, expansion trunks should be provided.

95 The use of the double bottom for fuel oil is open to several objections. The tanks being shallow and divided into a large number of compartments by the floors, keelsons, and intercostals of the ship, it is very difficult to fill them completely owing to the air trapped in the different compartments. It is also impossible to empty them

entirely and when a tank is only partly filled, trouble may be experienced in pumping out the oil when the ship is rolling. Again, if the ship has to go far North or South where the water is cold, the oil in the double bottoms congeals and difficulty is experienced in pumping. With many cargoes it is necessary to fill the compartments of the double bottom with water ballast when the oil is pumped out, and this means that at all times there will be a considerable quantity of water present in the oil. When the double bottom is used, it is necessary to provide settling tanks in or near the fireroom into which the oil is pumped from the ballast tanks before being pumped to the burners. The object of these settling tanks is to permit the removal of any water which may have found its way into the oil tanks, or which may be in the oil when it is loaded. These settling tanks should be of sufficient size to contain from 8 to 12 hours' supply. They should be in duplicate and fitted with steam coils for heating the oil, gage glasses to show the amount of water in the bottom of the tanks, and connections for the oil pumps and for pumps to draw off the water settled out from the oil.

OIL BURNING

96 The importance of proper furnace arrangements with means of controlling and directing the supply of air and fuel has been emphasized in previous discussions. In the case of marine installations it is not always possible to secure the best furnace arrangement. Nearly all the steamers in commission today are equipped with boilers of the internally-fired type. With the short cylindrical furnaces of comparatively small diameter used on these boilers, it is very difficult to secure the highest possible efficiency when burning oil. They are well adapted for the burning of coal.

97 For the burning of oil it is well known that an ample combustion space is needed. With a large combustion space, greater time is available and there is more opportunity for the oil particles to take up their requisite supply of air for combustion. Further, it is important, as has been pointed out, that the direction of the incoming air current should be such as to cause an intimate mixture of the air supply and oil particles. With the short cylindrical furnace these conditions are difficult of attainment. The air and fuel are admitted in sensibly parallel paths. The time during which fuel and air are in the furnace is very short. In consequence, complete combustion is difficult and is always delayed. With water-tube boilers, the furnace conditions are superior, and higher efficiencies have been shown.

98 In considering the installation of a fuel oil equipment, the subject is naturally divided into two parts, the first relating to the storage and handling of the fuel, and the second relating to arrangements for its combustion. Under the first, the safety of the ship and those on board her is the first consideration, and after this come convenience in handling fuel and accessibility of all the important parts; under the second, would be considered the system for burning oil, type of burners, and furnace arrangement.

ATOMIZATION

99 Three systems of atomization are in use in marine installations, namely, steam, air, and mechanical atomization. Of these, by far the greater number of installations in use on the Pacific coast are of the second order.

100 The use of steam for atomization is confined to vessels plying in inland waters, or on very short runs. Steam atomization is not suitable for vessels making voyages of any length because of the large amounts of fresh water necessary for boiler feed to make up the loss due to the steam used for atomizing. This feed water must be carried in tanks, thereby reducing the cargo capacity of the ship, or else made up by the use of evaporators, a very inefficient, expensive arrangement.

101 In air atomization the air is used at pressures ranging up to 60 lb. per sq. in., depending on the burner. By far the larger number of oil-burning outfits utilize pressures in the neighborhood of 20 lb. per sq. in., the air being supplied by a steam-driven compressor, or in the case of low pressures, a rotary blower.

102 The third system, mechanical atomization, quite generally known as the "Koerting" or "Meyer" system, although extensively used in Europe, has not yet been adopted to any extent on the Pacific. The atomization is effected by expelling the oil through a small orifice partly closed by a plug, on which is formed a spiral thread. The edges of the orifice are sharp and the spiral thread imparts to the stream of oil a rapid, whirling motion, causing the oil to break up into fine drops which leave the nozzle in a cone of atomized oil upon which the entering air currents impinge.

103 As has been said earlier in the discussion, few data are available showing the efficiency of marine oil installations. The amount of steam required for atomization will range from 2 per cent to 8 or 9 per cent, depending on the type of burner and the intelligence with which it is operated.

104 For air atomization and with air pressures of from 20 to 30 lb. per sq. in., from 6 to 10 cu. ft. of air per minute per pound of oil burned will be required. For air atomization with low pressures, such as can be produced by a rotary blower, of which the Lassoe-Lovekin system as fitted to the steamships of the American Hawaiian Steamship Company is an example, the amount of air required for atomization is not known. The air pressure used is $1\frac{1}{4}$ lb., and all the air is heated by the Howden system. The oil is heated to 175 deg. fahr.

105 A few installations of the third, or mechanical system, have been made on the coast, but no information is available as to the efficiency obtained. In the opinion of the writer, the mechanical system has not received the attention which it deserves from our local engineers. It should be efficient and its simplicity is certainly a recommendation. I am informed by the agent for the Koerting company that the oil may be handled by the ordinary pumps installed for that purpose, with other systems, but that an additional heater is necessary as the temperature of oil at the burner should be from 240 to 260 deg. It is further stated that as long as the pressure of oil at the burner tip is maintained at more than 40 lb. per sq. in., there will be no carbonization in the heating and pipe system, nor in the burner. It is said that 100 lb. per sq. in. at the burner is the most desirable pressure for this system. The makers state that to operate the pumps and supply the heat to the oil necessary with this system takes from $\frac{3}{4}$ to 1 per cent of the steam evaporated.

106 Some large tank steamers fitted with the mechanical system are said to operate very satisfactorily. The results obtained show that the consumption of oil is from 1.09 to 1.37 lb. of oil per i.h.p-hr. for all purposes. These vessels develop about 1900 h.p.

PRODUCER GAS FROM CRUDE OIL

By E. C. Jones, San Francisco, Cal.

Member of the Society

107 The subject of producer gas from crude petroleum or its products is not enough crystallized to enable the economy to be determined exactly. California, with its immense deposits of petroleum, is the natural and logical field for the exploitation and industrial use of oil producer gas. It is to be deplored that such an important subject was first considered by men not conversant with the manufacture

of oil gas, and in casting about for apparatus to make producer gas from oil they naturally gravitated to the old familiar methods of retorting the oil, and any improvements that grew out of these methods seem to have retained the objectionable features of the retort system. Briefly stated, these objections consist of shutdowns for the purpose of removing coke and frequently for burning out accumulated soot and lamp black. A typical analysis of gas made in this way is: CO_2, 4.5; CO, 7.4; O_2, 0.4; CH_4, 12.0; H_2, 3.1; N_2, 71.9; B.t.u. per cu. ft., 172; claimed thermal efficiency, 39 to 62 per cent; operating thermal efficiency, 55 per cent.

108 This gas has been applied to the operation of small gas-engine units up to and including 100 h.p. Owing to the abundance of petroleum in California, it has superseded all crude material in the manufacture of illuminating gas. To attain its present degree of perfection, elaborate experiments were performed for the purpose of changing the chemical and physical condition of the gas to best adapt it to modern domestic and industrial gas appliances.

109 The first oil gas manufactured on a large scale in California had the following analysis:

	Per Cent
Heavy hydrocarbons	6.2
Marsh gas	25.6
Hydrogen	62.4
Carbonic oxide	3.0
Carbon dioxide	0.2
Oxygen	0.4
Residual nitrogen	2.2
	100.00
Specific gravity	0.303

110 By improvements in apparatus and refinements of operation the hydrogen content of the gas has been reduced to less than 40 per cent; the marsh gas has been increased to 34 per cent; the carbonic oxide has been increased to 9 per cent; the specific gravity to 0.485, and the B.t.u. from 624 to 680 per cu. ft. The oil-gas generators used at present for manufacturing illuminating gas are so elastic in their operation that any of them can be immediately adapted to the manufacture of producer gas from oil.

111 The writer has carried on a series of experiments with oil producer gas, using a large generating unit and the only change in equipment was the use of compressed air at from 35 to 40 lb. pressure for the injection of oil and to assist in the partial combustion of

the oil. During these experiments producer gas was made having a thermal value as low as 103 B.t.u. per cu. ft., and as high as 482 B.t.u. per cu. ft. Unfortunately no ready means was at hand for measuring the quantity of gas and the amount of oil used to produce 1000 cu. ft. A typical analysis of this gas is as follows:

	Per Cent
Carbon dioxide	4
Heavy hydrocarbon	2 or less
Oxygen	1
Carbonic oxide	10
Hydrogen	5
Marsh gas	8
Nitrogen	70
	100

112 This gas has a calorific value of 160 B.t.u. per cu. ft. To use this gas successfully in gas engines it is necessary that it shall be thoroughly cleansed by efficient scrubbing, and that it shall be uniform in calorific value and chemical constituents. This last can be readily accomplished in the operation of oil-gas generators, used as producers, by careful measurement of the oil used, and of the air supplied for its partial combustion, and by the maintenance of a fairly constant temperature in the generator through observations of the color in the checker brickwork.

113 Making producer gas from oil in the ordinary oil-gas generator has many advantages over any special process. The gas can be made in very large quantities and the amount made can be easily regulated to the needs. The operation is continuous and without interruptions for cleaning. This is essential in the manufacture of producer gas for power purposes. In any other known process the interruptions are not at stated intervals, but occur when the producer refuses to work, owing to the clogging of its parts by coke or lamp black. Oil-gas generators have no easily destructible parts as the lining and checker brick are constructed with a view to resisting high temperatures and although a much higher temperature is desirable in making producer gas than that employed in making oil gas, the checker brick is not seriously affected by the high temperature. In the decomposition of oil, in the presence of air, there is a complete disposition of all the gas making constituents of the oil, so that producer gas can be made without a by-product of any kind. Any accumulation of carbon in the generator may be removed by adjusting the temperature and quantity of air supplied. This method of making

gas requires a small gas holder for momentary storage, and the process as at present understood could not be used as a suction gas producer. Producer gas made from oil and containing a small percentage of hydrogen possesses advantages over illuminating oil gas, inasmuch as it can be subjected to higher compression in gas engine cylinders, and with a gas of uniform analysis, uniform piston speed can be obtained.

DISCUSSION

W. H. Frost[1] mentioned the following as requisites in producing power gas from fuel oil:

a To produce the gas continuously in the quantity required by the engine, dispensing with storage.
b The gas to contain the maximum of carbon monoxide and the minimum of hydrogen, to obtain highest thermal efficiency and best regulation of engine.
c Gas to be practically even in quality to prevent shutdowns.
d Gas to be clean, as very minute particles of carbon accumulate and foul engine valves, necessitating shutdowns and cleaning.
e Apparatus to occupy relatively small space.
f Operation to be simple, readily understood by ordinary attendant and requiring minimum of attention.
g To dispense with offensive fumes and by-products.
h To reduce the necessary elements to the minimum.

During experiments of several years' duration, in making producer gas from oil, Mr. Frost had found the most difficult task of all was to separate the fine, almost microscopic particles of carbon from the gas, so that it would be perfectly clean and suited to any gas engine. Eventually, a centrifugal scrubber of simple design was devised, which absolutely cleans the gas, so much so that fine white muslin held for one or two minutes over a 1-in. outlet with gas under 2-in. water pressure, sufficient to make a very long flame, shows no stain whatever and if kept there for 2 hours shows only a slight discoloration. The scrubber doing this is about 4 ft. in diameter by 5 ft. long, and cleans about 35,000 cu. ft. of gas per hour.

The fine dry carbon, free from tar, is collected from the wash water by any convenient means. In a plant of 500 h.p. it is not sufficient in quantity to require any special separating device, but is collected

[1]Pres., Natl. Oil Gas Producer Co., Los Angeles, Cal

in a skimming tank in the usual manner. This fine clean carbon has many valuable uses, particularly in paint for metal or wood.

The economy, of course, varies with the engine, but actual runs on plants of 100 to 500 h.p. show that a guarantee of $1\frac{1}{8}$ lb. of oil per b.h.p. is perfectly safe.

As to sizes of producers, with a circular chamber less than 18 in. in diameter by 4 ft. long, or about 7 cu. ft. capacity, I have made the gas required to operate a 150-h.p. engine full load, but prefer larger capacity, at least for the present.

Mr. Jones in his paper on producer gas from crude oil states: "The operation is continuous, and without interruptions for cleaning. This is essential in the manufacture of producer gas for power purposes." Quoting again, "Any accumulation of carbon in the generator may be removed by adjusting the temperature and quantity of air supplied." This indicates that there is a necessity for cleaning but that a change in proportion of air, with consequent change in temperature (probably more air raises temperature), causes the accumulated deposits to be consumed. The writer's experience had been that any change in temperature, or proportion of elements, changed the quality of the gas and caused shutdown of the engine. The change mentioned would probably make the gas leaner.

The only way to avoid serious engine troubles would be to provide sufficient storage so that the changes of gas values would be very slow and gradual and even then the load must never be greater than the leanest gas will carry, and the engine requires close attention and skilful handling. Even if the leanest gas can be kept up to 160 B.t.u., it will require 37,500 cu. ft. per hr. for a 500-h.p. load (12,000 B.t.u. per b.h.p), making an enormous storage capacity necessary to anywhere near the average of the quality of the gas.

The statement of the necessity of changes also shows that this is not a continuous process producing a uniform gas, but one producing a continuous supply of gases of varying values, whereas a continuous supply of a practically uniform gas is absolutely essential in engine operation.

C. R. Weymouth. If we assume that the range of fuel oil ordinarily varies between the limits of 12 and 18 deg. Baumé, we find, in accordance with Table 3 of Professor LeConte's paper an increase of 2.18 per cent in the total heat units per barrel, in favor of the heavier oil. So long as the barrel is the unit in the purchase of oil, power plant operators cannot ignore this comparison.

Since the reading of these papers the writer has endeavored to collect data in an attempt to establish a relation between the specific gravity of California crude oil, Baumé scale, and the percentage of hydrogen shown by its ultimate analysis in order to compute a table showing the available heat per pound and per barrel of crude oil of varying gravities, correcting for loss due to latent heat of steam formed by the combustion of hydrogen. A curve was plotted showing the hydrogen content for a number of oils of different gravities, Edmund O'Neill, professor of chemistry at the University of California, having furnished a large part of the data.

It was found that there is no exact relationship between the specific gravity of crude oil and its hydrogen content, although there is a general tendency toward an increase in hydrogen content, with the lighter oils.

Examination of the ultimate analysis and calorimeter tests of a number of California crude oils indicates the rather startling fact that it is possible in oils having practically the same total quantity of inert constituents, to have a variation in both hydrogen and carbon contents with practically no variation in the calorific value of the oil.

The inevitable conclusions that the calorific value of crude oil does not correspond to the heat of combustion of its elemental constituents, and that accepted formulae for calculating the total heat of fuels are not applicable to California crude oils, are borne out by the fact that the calculated calorific value was, in one instance, 8.7 per cent greater than obtained from a calorimeter test. This apparent anomaly is no doubt explained by the fact that crude oil is an admixture of various hydrocarbons, in the formation of which the heat required is not available in the further combustion as crude oil.

From Table 3 the writer calculated the variation in hydrogen due to variation in heat units, according to Favre and Silberman's formulae, the result being an increase of 1.263 per cent hydrogen for 10 deg. increase Baumé. From the foregoing it is evident that this relationship cannot be regarded as more than an approximation. As a matter of interest, a diagram was plotted and a line drawn having its slope in accordance with this, and located as an average with respect to the plotted points. On the basis of this line and correcting for latent heat of steam only, the available heat per barrel of oil is 2.32 per cent greater for 12 than for 18 deg. Baumé oil. If further corrections be made for the added stack losses at a temperature of 400 deg. fahr. due to the greater air required for combustion,

then 12 deg. Baumé oil has an advantage over 18 deg. Baumé oil of 2.37 per cent.

It should be evident from the wide scattering of the hydrogen points that the comparison of any two oils under consideration should be made with respect to their individual analyses, and not with respect to the average line mentioned. For example, from the plotted points a hydrogen variation of 1.6 per cent is noticeable with practically no variation on the Baumé scale, corresponding to which the stated increase of hydrogen involves a commercial loss, due to its lower available heat, and increased chimney losses of approximately 1 per cent. Taking into consideration this loss and the greater total heat units of heavy oils, the extreme variation in the commercial value of anhydrous crude oils, between the limits of 12 and 18 deg. Baumé, appears to be not greater than 3 per cent.

As a result of all the tests made by Chas. C. Moore & Company Engineers, and the Babcock & Wilcox Company, with California crude oils, it is not apparent that the above conclusions will be materially modified by a consideration of the atomizing agent, when oils are heated to the proper temperature before firing.

It should be noted that Table 3 is intended to represent a general average, the line being drawn through plotted points considerably scattered. Calorimeter determinations will still be necessary as the specific gravity of an oil is not an accurate index of its calorific value.

Mr. Dunn has fairly represented current practice in the design of chimneys for oil-fired boilers at sea level, for moderate sized plants.

About eight years ago, the writer established, for Chas. C. Moore & Company, a rule for oil-burning chimneys, as set forth in Mr. Dunn's paper, and while such a basis for the selection of chimneys has kept them out of difficulties, recent investigations indicate the desirability of a radical departure from this rule in certain special instances.

For the benefit of those who contemplate using Kent's table of chimney capacities as a basis of selection of chimneys for oil-burning plants, the writer would state that Mr. Dunn's explanation of the reasons for adducing the rule given, is hardly correct. The weight of chimney gases per boiler horsepower, when burning coal, is not twice that when burning oil. It is possible to use chimneys of smaller area for oil burning than for coal burning, by reason of the much lower draft required for the former, permitting higher velocities in the chimney, and a greater percentage of draft loss in the chimney proper.

Mr. Kent probably never intended the widespread usage accorded his table of chimney capacities, and before adopting this for too general usage, engineers would do well to inquire the basis of development and the significance of the figures given.

According to Kent's table, a chimney 48 in. in diameter by 100 ft. in height, is rated at 348 commercial h.p. According to Mr. Dunn's paper, this chimney would develop double that amount, or practically 700 boiler h.p. The questions arise, should this chimney be connected to boilers aggregating 700 h.p. based on the normal rating, will it merely carry boilers when operating at rating, or does it provide a margin for overload, and if so, to what extent; and were it desirable, in such a plant, to operate both boilers temporarily at 100 per cent overload, developing 1400 boiler h.p., would it be necessary to install a chimney of double the sectional area? The writer no longer uses Kent's table for selecting chimneys for oil-fired boilers, and it is not possible to obtain correct ratings for oil chimneys by applying a fixed ratio to the capacity set forth in it.

With economical firing, and with properly designed furnaces, the draft in the third pass, required to burn oil in a Babcock & Wilcox boiler of, say 250-h.p. rating, is about 0.1 in. If this boiler be connected to a chimney 100 ft. in height, giving a working draft at the base of the chimney of, say 0.5 in., it is evident that this surplus draft is capable of flooding the boiler with an excess of air, if the draft be improperly regulated. Such an operation could easily result in such a large excess of air for combustion as to require the burning of 10 per cent excess oil, merely to heat to the stack temperature the air so admitted.

If for the given single boiler the chimney height be reduced approximately to 35 ft., the boiler outlet and breaching being of ample area, and the stack being direct connected to the boiler, the chimney would then produce a maximum draft practically equivalent to that required for the operation of the boiler at rating; and with boiler damper and ashpit doors wide open, it would be impossible for the most careless firemen to flood the furnaces with any material excess of air. The writer does not recommend that all chimney heights be reduced to 35 ft., but this illustration serves to indicate an extent of safeguarding the fuel economy of the boiler room, not possible by any other simple means.

In an office building in San Francisco, it was found necessary, by reason of the height of the building, to install a chimney extending more than 200 ft. above the boiler room floor line. On one occasion

the analysis of flue gases, in connection with this chimney, indicated 200 per cent air over actual requirements, and a certain fuel loss, as compared with economical firing, of at least 20 per cent. While, in this instance, it was impossible to reduce the height of the chimney, the diameter could have been decreased to such an extent, that the available draft at the base would have been reduced by the chimney friction to an amount more nearly in keeping with the boiler requirements.

As illustrating the overloading and abuse of boilers possible with excessive drafts, the writer has in mind a Stirling boiler having a connected chimney of generous diameter, 80 ft. in height. The damper lever had been disconnected, and the boiler fired with both rear damper and ash doors wide open. The fireman was instructed in the regulation of dampers and ash doors, but failed to observe these instructions. Subsequent tests indicated overloads of several hours' duration, of 85 per cent, and momentary overloads of even greater extent. There were other boilers in service, having less draft power, which were not overloaded to this extent. In the plant in question, all boilers should have been operated practically at rating on a nearly uniform factory load. Obviously the overloading of this particular boiler led to a considerable fuel loss. To obviate these difficulties the writer has recently recommended a 40-ft. reduction in the height of this chimney. A greater reduction would be possible from the boiler standpoint, but is in this instance limited by height of building.

When operating boilers under variable load, a careless fireman is liable to discover that the steam gage reads 5 or 6 lb. low and rapidly falling. Through habit, he has learned that this loss can be overcome by a few moments of heavy firing. During such periods, the boiler is frequently fired at from 200 per cent to 300 per cent rating and boiler tube renewals are many times chargeable to this cause, not to mention the resulting excessive oil consumption. While a certain excess draft is necessary for proper regulation of boiler plants on variable load, it is self-evident that it can lead only to severe abuse of boilers in the hands of ordinary firemen.

Prior to the burning of oil in Pacific coast plants, chimneys were proportioned for coal-burning practice, and in many instances the subsequent failure to obtain favorable economy with oil fuel in everyday work was attributable to tall chimneys. In a number of those stations it was later found possible to increase largely the number of boilers connected to one chimney, the station economy was improved

without apparent effort of the firemen, since the increased volume of gases passing through the flues and chimney reduced the draft available at the boiler outlet, and at once limited the extent of flooding the furnaces with excess air, formerly possible with an excessive draft.

It has been found by experiment that the draft necessary for the combustion of oil varies largely with the economy of firing, and in one series of tests, operating the same boiler at approximately rated capacity, the draft varied from less than 0.1 in. with economical firing, to nearly 0.5 in. with a large excess of air; also that the draft necessary to burn oil increases rapidly with the rate of overload on boiler. It therefore becomes necessary, in the selection of chimneys for a given plant, to decide both the maximum capacity to which it is desirable momentarily to force boilers, and the excess air supply over the most favorable firing conditions, to be regarded as the limit of safety in everyday practice. The installation of fuel economizers, excessive breeching resistances, etc., will also influence the final result. These considerations determine the draft necessary at the base of chimney. There is a minimum height of chimney which will produce the desired draft, and should it become necessary to increase the chimney height over and above the necessary minimum, the diameter of the chimney, for economical firing, should be modified to absorb by chimney and breeching friction the excess draft thus produced. The provision for a later increase of boilers on a given chimney will of course modify these conditions.

There are many factory, pumping and heating plants where the load on the boilers in service is practically uniform and rarely exceeds rating. In certain of these plants, not operated by skilful firemen, it would be possible, building conditions permitting, to reduce the chimney height materially with an increase in fuel economy.

In reply to a query by Mr. Hunt, by controlling the firing of boilers by variation of oil pressure from a central point, all burners being connected to a common oil main, it is possible to indicate the rate of load on the individual boilers by pressure gages connected to the oil-burner branch line. At the Redondo plant we controlled the firing of boilers from a central point, the individual oil-burner regulating valves being wide open or nearly so. The pressure gage read about 20 lb. when the boilers were fired at rating, about 10 lb. at half load, and about 30 lb. when at 50 per cent overload. When used in this manner the pressure gage forms a reliable index for gaging the load on boilers, at least for the purpose of the firemen, and it is about as cheap an instrument as can be secured for the work.

W. F. DURAND. In connection with Mr. Dunn's paper on chimney capacities, the accompanying formulae may be submitted as representing conservative practice under what may be denominated approximately normal operating conditions. The factors which enter into the problem of chimney draft and capacity are so numerous, their relations so obscure and values so difficult of numerical determination, that of necessity all chimney formulae are empirical, that is, they are simply practice expressed in algebraic form. The literature of engineering abounds in chimney formulae, but inasmuch as most of them relate to coal-burning practice, I have ventured to add one more to the list, intended to refer strictly to oil-burning practice.

While, as noted, all chimney formulae are empirical in character, yet they may to some extent be made rational in form and the more nearly such a condition is realized the better will the formula take care of variations in the operative conditions, outside the immediate range from which the empirical factors are drawn.

The general character of a chimney formula on rational lines may be readily developed, as follows: The function of draft involves the pushing of air into the furnace and the pushing of the gases of combustion through the tube spaces and then up the stack to the outer air. This involves the work of giving kinetic energy to a body of air and gas and of moving it against a resistance. In general, such work will depend on the quantity of gas moved and on the square of the velocity. This work must be supplied by the draft head or difference of pressure between the external air and minimum pressure within the boiler, and per pound the work will be measured by the difference of pressure in pounds per square foot multiplied by the volume of one pound of gas in cubic feet. We have thus in general the relation:

$$u^2 \sim VP$$

where

u = velocity
V = volume per lb.
P = draft head measured in lb. per sq. ft.

Again, let

B = boiler h.p.
w = lb. of gas per boiler h.p.
A = area of chimney

Then

BwV = total volume of gas = Au

But $u \sim \sqrt{VP}$, and P the draft head will depend in general on the height of the stack and on the stack temperature. According to the simple theory of stack draft, such head is measured by the difference in weight between a column of normal air and a column of hot gas, each of the height of the chimney.

In any event, we may put

$$P \sim hx$$

where

h = height of chimney
x = a factor or term depending on the temperature

We have then

$$u \sim \sqrt{Vhx}$$

and hence

$$A \sqrt{Vhx} \sim BwV$$

whence

$$A \sqrt{h} \sim Bw \sqrt{\frac{V}{x}}$$

In words, the product of the area by the square root of the height should follow closely the product of the number of pounds of gas by the square root of the volume per pound and divided by the square root of the function x.

But the volume of 1 lb. of gas V varies directly as the absolute temperature. This of course varies widely between the furnace and the top of the stack. The temperature at the base of the stack is usually taken, however, as primarily related to the value of the draft function x, and for convenience we may use the same temperature as related in a general way to the value of V. Also w, the pounds of gas per boiler horsepower, will vary inversely as the boiler efficiency.

Let e denote the efficiency and Ta and Ts the absolute temperatures of the air and of the stack. Then we may write

$$A \sqrt{h} = \frac{B}{Qe} \sqrt{\frac{Ts}{x}}$$

where Q is the general constant or factor.

The function x takes the form $\frac{P}{Ta} - \frac{R}{Ts}$, where P and R are constants. Where draft head d is expressed in inches of water and

the height of the chimney in feet, this takes the form

$$d = h\left(\frac{7.6}{Ta} - \frac{7.9}{Ts}\right)$$

If 80 deg. fahr. is taken as a fair upper temperature of the air, this reduces to

$$d = h\left(0.1407 - \frac{7.9}{Ts}\right)$$

The entire $\sqrt{\frac{Ts}{x}}$ is therefore a function of the temperature and its value for a range of values of Ts is given in Table 8.

It now remains only to find a value for the numerical constant Q. This has been done by using the results of practice and thus checking

TABLE 8 VALUE OF TEMPERATURE FUNCTIONS

t	y	t	y
400	4.19	560	4.02
420	4.16	580	4.01
440	4.12	600	4.00
460	4.10	620	4.00
480	4.07	640	4.00
500	4.05	660	4.00
520	4.04	680	4.00
540	4.03	700	4.00

the formula against the indications of experience.

In this manner, we find as follows:

$$(A - a)\sqrt{h} = \frac{By}{Qe}$$

where

A = area of chimney in sq. ft.
a = small constant.
h = height in ft.
B = boiler h.p. actually developed, which equals number of pounds of total evaporation per hour reduced to conditions from and at 212 deg., divided by 34.5
e = boiler efficiency
y = temperature factor from the table
Q = coefficient

Values of a and Q are as follows: For large chimneys or from 300 boiler h.p. up

$$a = 2.5$$
$$Q = 54$$

For small chimneys or from 500 h.p. down

$$a = 0.6$$
$$Q = 40$$

In choosing the values of the constant Q the following special conditions are assumed:

Heat value of oil fuel	about 18,500 B.t.u.
Temperature of external air	80 deg. fahr.
Excess of air in furnace	about 100 per cent

Boilers working at rate of output not greatly exceeding rated load.

The particular numerical factors in the formula are perhaps not so important. I dare say further investigation, undoubtedly the results of Mr. Weymouth's data, might change the numerical factors. I believe, however, that the form of the formula may possibly be of some aid in passing from one set of conditions to another, or interpolating between various sets of conditions; and that is about all any engineering formula can hope to do.

In connection with the paper by Mr. Hunt on Atomization, I have been interested in estimating the work equivalent of this process. The nozzle of a burner is a more or less effective device for transforming energy of steam into jet energy, exactly in the same manner as does the nozzle of a steam turbine. The atomization of the oil and its projection at high velocity into the furnace involve the expenditure of work, and this work must be derived from the steam by way of the nozzle.

One-half lb. of steam per lb. of oil is perhaps a fairly representative figure for the amount of steam required. This amount of steam used with any reasonable nozzle efficiency and under the conditions of, say 90 lb. initial pressure absolute and 15 terminal, should develop from 35,000 to 40,000 ft-lb. of energy, thus expended in work on the oil. This figure is impressive. It is probable that due to wire drawing and inefficiency in the nozzle the amount of work actually utilized is less than this figure. In any event, however, the price paid for the preparation and introduction of the oil into the furnace is a very heavy one, and the question not unnaturally arises as to whether or not this can be the ultimate method. May not some method be developed, mechanical or otherwise, which shall enable us to do the

necessary amount of work on the oil without the heavy expenditure involved in the present systems of steam or air atomization. Mechanical atomization seems to present some possibilities and it may be that the future development may lie along this path. In any event, I feel that as engineers we should entertain a feeling of profound discontent with our present methods of preparing and introducing the oil into the furnace, and that we should not rest until either some less costly method is developed, or every possible method and expedient has been given a thorough trial.

GUY L. BAYLEY.[1] In order to get some information as to the efficiency of the furnaces in the station of the Municipal Light & Power Company plant, on Stevenson Street, we installed an Orsat apparatus in the fire room. The four boilers are of Stirling make, and the samples were taken about a foot below the damper. The determinations were of great value at the start in educating the firemen as to the amount of air required. All of our men were experienced in the burning of fuel oil, but they were firing with from 80 to over 150 per cent excess air. By shutting down on the draft and making frequent determinations with the Orsat apparatus until a high per cent of CO_2 was obtained, the firemen were taught how a fire should appear with the minimum amount of air for complete combustion. We were unable to get much better than 12 per cent CO_2, as above this point the fire was liable to produce smoke, which would not long be tolerated in the neighborhood of our plant.

The second month showed an increase of 19.3 per cent in the kw-hr. per bbl. of oil obtained the month before installing the Orsat. During this period the load factor on the turbine increased from 43 to 53.5 per cent, which would account for some of the increase in economy, but hardly for the amount obtained.

THOS. MORRIN. The use of crude oil as fuel has been developed along the lines of unlimited extravagance in the vaporizing process. Until recently steam jets have been used in land plants at full boiler pressure for vaporizing oil at the burner tip, because of its convenience and low cost of application. At the same time it is perhaps the most expensive method for accomplishing this object. I believe that improvement in these conditions can be brought about only by a burner mechanism that will require air only at an extremely low pressure

[1] Mgr., Municipal Light & Power Co., San Francisco, Cal.

which may be furnished at a minimum cost so that its most extravagant use cannot go beyond a certain fixed range.

I have in mind mechanical burners of the turbine fan-wheel type rotating at a high speed and delivering the oil from the periphery of the burner at a velocity of approximately 7100 ft. per min., equal to an air pressure of approximately $1\frac{1}{2}$ oz. per sq. in.

In tests of the quantity of oil and the percentage of power used to burn a fixed quantity it was noted that the energy expended was less than 1 per cent. In fact, the figures showed an actual consumption of energy amounting to but $\frac{6}{10}$ of 1 per cent of the actual heat results of the oil burned.

I refer to this scheme of vaporizing oil because I am sure that oil will be used more and more for fuel purposes if we convince the consumer that it is unnecessary to install a steam burner or an elaborate mechanical equipment. A simple centrifugal blower and a rotary gear pump driven by one electric motor that any unskilled person may use and operate is a possible equipment.

E. J. Dyer[1] stated that he had had access to records of performances of several steamers running on the coast with average coal for an entire year. One of these vessels averaged 2.1 lb. of coal per i.h.p-hr., and the other 2.5, and both had triple-expansion engines larger than the lower limit of size given by Mr. Hopps. The better record showed about 70 per cent more weight of coal and the poorer 100 per cent more than the $1\frac{1}{4}$ lb. average for oil quoted by Mr. Hopps.

In regards to stacks, if the correct height were 80 to 100 ft., how could the fact be accounted for that a boiler plant of 600 h.p. capacity with stacks only 51 ft. above the floor line, recently installed by the speaker, developed an average of 170 per cent of rating with peaks running up to 200 per cent on a five-day test? Not only this, but the height is such that the firemen have to be watched to prevent the admission of an excess of air. Moreover, in the same building a recent storm has entirely destroyed a number of stacks on some return tubular boilers down to the roof line, and there is not the slightest difficulty in running these boilers up to capacity with the stacks in this condition. In this case, the height of stacks could not be more than about 25 ft. above the floor. If all the overload capacity consistent with safety could be secured and the firemen still get an excess

[1] Engineer of Manufacturing Department of the Union Oil Company, San Francisco, Cal.

of air if not watched, manifestly 50-ft. stacks are too high. At another plant of 1200 h.p. in four units, the stacks are 72 ft. high. These are made higher than would be ordinarily necessary on account of the configuration of the surroundings. With these stacks, the boilers can be operated at any reasonable overload and if the fireman is not watched he will get his CO_2 as low as 2 per cent. These stacks are, therefore, too high, except perhaps under conditions which might bring to bear the adverse influence of the winds passing over the surrounding hills. It is manifestly unsafe to generalize on either the height or diameter of stacks without knowing all of the conditions. The draft resistance differs in different types of boilers; the amount of overload desired has an influence on the height and area; the draft resistance in the boiler increases with overload; conditions which obtain at sea-level are not present at altitude; the direction of prevailing winds as regards the situation of the plant with reference to the configuration of the surroundings is of consequence; there is a great difference whether each boiler is equipped with an individual stack, one stack per battery of two, or one stack serving several, with connecting breeching; every turn in a breeching introduces a draft loss; the frictional resistance and the cooling effect of economizers must be taken into consideration; for construction purposes there is an economical ratio of height to diameter; the price of fuel oil is a consideration, etc. So many factors enter into the design and proportions of stacks that the subject is a complicated one, and worthy of more extensive exploitation.

As to the use of compressed air instead of steam for atomizing, a special occasion for using it in place of steam is probably offered by plants where exhaust steam does not exist in sufficient quantities for heating purposes within the limits of temperature obtainable by steam at or near atmospheric pressure. A plant of 2000 boiler h.p. located in Seattle and used entirely for a district steam heating system would seem to present a possibility for this practice. Industrial operations involving the use of large quantities of heat at relatively low temperatures, of which there are a number of examples, might also be similarly situated.

G. H. Marx said that in 1904 he visited the laboratories at Munich, and was shown the engine on which Professor Shröder had performed his experiments, by his assistant, who assured him that they had used California crude petroleum in some of their tests, and that the petroleum had worked very satisfactorily, and left no residue in the cylinder.

HOWARD STILLMAN spoke of the operation of Diesel engines, relating experiences in the shops of the Southern Pacific Company at Ogden, Utah, and at Tucson, Ariz. He recounted how attempts had been made to operate the engine at Ogden with California crude oil, which were without success because it was frequently stopping and tying up the shop. A lighter distillate was then used which worked very well and was exceedingly economical, but even then, the engine was not reliable. The shop could not be kept going without resorting to the old steam engine. There were records of three weeks at a time when the performance was remarkable and then would come a period of, say, two weeks when the Diesel Engine was hung up for repairs and the steam engine had to be used.

At Tucson the engine was maintained longer, covering a period of three years, and the same attempts were made to use the crude oil or a lower gravity oil, but the engine was intermittent in its action. Finally, a Texas oil of lighter gravity than the California oil was adopted and the performance of the engine was fairly good, although it had its difficulties. When it was working it was a splendid machine. One attempt was made to dismantle the steam engine but fortunately this had not been completely done, when its services were required because of trouble with the Diesel motor. The motor was finally abandoned as it was found there was no net economy in operation. It could not be prophesied when it was going to stop and the steam engine was finally put into regular service again. Delays in operation of lineshafting in machine shop during working hours were expensive.

R. W. FENN said that he had been in Chili and had found Diesel engines quite common there and inquired particularly about their operation from engineers in charge of them. They pronounced them ideal, but were using a Peruvian crude oil of 34 deg. gravity.

A. F. L. BELL reported trouble in the operation of a 1000-h.p. Diesel engine installed at the United Verdi Mine. The engine had been bought without giving attention to where the oil was to be obtained and experiments were now being tried with samples of oil and distillate from different parts of the California oil fields. He thought that the California oil must have a carbonizing effect, causing this type of engine to stop at intervals.

THOS. MORRIN said that he had had occasion a few years previously

to inspect all of the carburetors made in the vicinity and that he found by using the carburetors in duplicate, so that if one fouled a change could be made to the other, and by feeding water into the inoperative one while it was still hot, the deposit of dirt and carbon would slough off easily and it could be readily cleaned. The carburetor must be so constructed that it could be easily opened and tried out. If the oil was very dirty it might be necessary to have the carburetors in triplicate. No difficulty had been found in operating when the carburetors were cleaned with water or steam.

No. 1308

OIL FUEL FOR STEAM BOILERS

By B. R. T. Collins, Boston, Mass.

Member of the Society

1 In order to make comparisons between the calorific value and other properties of crude oil, fuel oil and any particular coal, Table 1 is given, with the authority therefor in each case.

TABLE 1 PROPERTIES OF CRUDE AND FUEL OIL

Oil	Field	Carbon	Hydrogen	Sulphur	Oxygen	Specific Gravity	Flash	Fire	B.t.u.	Authority
Crude	Sour Lake, Tex.	.....	.....	.....	.....	0.9266	198	.....	18460	Prof. A. C. Scott,
Crude	Beaumont, Tex.	.....	.....	.....	.....	0.9179	.....	.....	18500	Univ. of Texas
Crude	Beaumont, Tex.	84.6	10.9	1.63	2.87	0.9240	180	200	19060	U. S. Naval Liquid
Fuel	Beaumont, Tex.	83.3	12.4	0.50	3.83	0.9260	216	240	19481	Fuel Board
Crude	Whittier, Cal....	.....	.....	.....	.....	0.9416	.....	.....	18513	Prof. W. C. Blasdale, Univ. of Cal.

ADVANTAGES AND DISADVANTAGES OF OIL FUEL

2 The advantages of oil fuel may be summarized as follows:

a Calorific value per pound 30 per cent higher than that of high-grade coal, a less weight of oil being required to give the same heating effect.

b Space required for storage of oil is less than that for an equal weight of coal. Fifty per cent more heating value can be stored in the same cubic volume and at greater distance from the boilers without extra expense.

In abstract form. Presented at the Boston meeting, April 1911, of The American Society of Mechanical Engineers. Complete paper and discussion published in The Journal, August 1911, p. 931.

c Oil does not deteriorate by storage, as coal does to a greater or less degree, but maintains its heat value indefinitely in ordinary ventilated storage tanks.

d Lower temperature in boiler room.

e Area of stack 60 per cent of that required for coal for equal boiler capacity, thus enabling a plant having insufficient draft with coal to have an excess amount with oil, a change from coal to oil for fuel making installation of additional stack capacity unnecessary.

f Less heat lost up the stack, owing to cleaner condition of tubes and to smaller amount of air which has to pass through furnace for a given calorific capacity of fuel.

g Higher efficiency due to (1) more perfect combustion with less excess air, (2) more equal distribution of heat in combustion chamber, as doors do not have to be opened, and (3) small amount of soot deposited on the tubes.

h Increase in capacity of 35 per cent to 50 per cent over coal, depending on grade of coal and draft conditions with coal.

i Heat is easier on the metal surfaces, being more evenly diffused over the entire heating surface of the boiler.

j Ease with which fire can be regulated from a low to a most intense heat in a short time or entirely extinguished instantly in case of emergency, such as water dropping out of sight in gage glass, and quickly relighted when the emergency is over. In less than half an hour a boiler can be brought up to 150 lb. steam pressure from cold water, if necessary. By means of an automatic regulator varying the pressure of oil and steam or air to the burners or atomizers, the steam pressure can be maintained within 5 lb. total variation, with sudden changes in load amounting to 50 per cent and over.

k Smoke can be entirely eliminated.

l No cleaning of fires, thus boilers can maintain their maximum capacity continuously, if necessary.

m Much lower cost for handling oil, as it runs by gravity or is pumped into and out of storage to the boilers.

n Absence of coal dust and ashes, thus enabling everything in the boiler room to be kept clean; therefore less wear and tear on pumps or other machinery. No expense for handling and removing ashes.

o No firing tools used, consequently no damage to furnace linings from this source. No clinkers to be removed from grate bars or furnace side walls.

p Less shrinkage and loss in handling oil fuel than in handling coal.

q Great saving of labor of all kinds: firemen, coal passers, ash wheelers, tube blowers, etc.

3 The disadvantages of oil fuel are:

a Low flash point. Fuel oil should have a flash point not lower than 140 deg. fahr., and with oil of this quality, handled by men of ordinary intelligence and common sense, there is practically no more danger than with coal.

b The ordinary underwriters' or city requirements specify that storage tanks for fuel oil be located underground and at least 30 ft. from the nearest building. This can generally be complied with in the case of the power plant of the average manufacturing concern, but in the case of a plant in the congested districts of a city it is likely to be prohibitive.

c With boilers using feedwater of considerable scale-making qualities, the cost of repairs is likely to be increased by changing to oil, owing to the intense temperature developed in the furnace. However, with a proper setting for burning oil, repairs due to overheated tubes or surfaces should be less than with coal, unless the feed water is very bad.

PRINCIPLES INVOLVED IN EFFICIENT OIL BURNING

4 The requirements for the perfect combustion of liquid fuel are as follows: Reduction to a fine spray or complete atomization; bringing it into contact with the proper amount of air; mixture of oil spray and air burned in a furnace of a refractory material with room enough to complete combustion before the gases come in contact with the boiler heating surfaces.

5 The first condition is fulfilled by selecting a proper burner, and the remaining conditions can generally be obtained by making slight changes in, or additions to, the existing furnaces.

6 The question whether to use steam or air for atomizing the oil seems generally to have been decided in favor of steam, as experimental results show that it takes about the same amount of steam

to operate the air compressor as it does to atomize the oil at the burner and the additional investment and complication involved with greater possibility of interrupted service is avoided.

7 It is also easy to see that a flat fan-shaped flame presents a larger surface for heat radiation and uniform distribution of gases than a flame of any other shape, and at the same time requires a minimum number of burners per boiler.

8 Heating of the oil is an aid to economical combustion, and should take place as near the furnace as possible and be carried as high as safety permits, but not so high as to cause the oil to decompose and carbon to be deposited in the supply pipes. If preliminary heating is limited to the temperature of the flash point of the oil used, there can be no trouble from the above-mentioned causes.

9 In oil burning, although a certain amount of skill is required for hand adjustment of the burners to obtain the best results, still, with automatic regulation, the skill is reduced to a minimum, the principal work of the fireman being to see that the oil pump is kept in constant operation and that the burners do not become clogged with small particles of foreign matter, scale, etc., especially when the installation is new. Strainers of proper design, however, introduced on the suction line to the pump and also between the pump and the burner, will reduce this trouble to a minimum. Burners should be so installed that they can be easily disconnected from the piping and taken from the furnace for the removal of any foreign substance from their restricted orifices.

10 One of the most important questions in the combustion of liquid fuel is the regulation of the air supply in such a way as to obtain perfect combustion before the gases come in contact with the heating surfaces of the boiler. This can be done with an automatic damper regulator, although its adjustment is rather difficult. It is therefore usually accomplished by hand regulation of the damper when considerable variations in the load take place. This is supplemented by changing the position of the ash-pit doors, which are kept partly closed until a slight tendency to make smoke is noticed in the furnace when they are opened until this tendency disappears; or, better, by using an Orsat or continuous CO_2 gas analyzer to determine the position of damper and ash-pit doors which gives most complete combustion under certain constantly recurring conditions.

TYPES OF OIL BURNERS

11 Although thousands of patents have been granted for oil burners or atomizers two general subdivisions or five general classes, as designated by the U. S. Naval Liquid Fuel Board, cover practically all of the main features of construction.

12 The two general subdivisions are:

a Outside mixing. Oil and atomizing agent meet outside the burner.

b Inside mixing. Oil and atomizing agent meet inside the burner.

13 The five general classes included in Par. 12 are:

c Drooling. Oil oozes out onto steam or air jet.

d Atomizing. Oil is swept from orifice by steam or air jet.

e Chamber. Oil mingles with steam or air in body of burner and the mixture issuing from nozzle is broken into minute particles by the expansion of the steam.

f Injector. Similar in principle to boiler-feeding injector.

g Mechanical spraying. Effected by mechanical means without the use of atomizing agents, such as steam or compressed air.

14 The important features which should be embodied in all burners are: Easy method of installation, construction that will allow quick inspection, easy removal of all foreign material which may clog the burner at any point, and rapid and cheap renewal of anyparts which are subject to wear.

METHOD OF INSTALLATION

15 In spite of the various principles involved in burner construction, the success of an oil fuel installation depends not so much on the type of burner or atomizer used as on the method of its installation, and the intelligence with which it is operated after the installation is made.

16 To conform with the underwriters' requirements, storage tanks above the surface of the ground should be placed at least 200 ft. from inflammable property, and the top of the tanks should be located below the level of the lowest pipe used in connection with the apparatus. When the tanks are located underground, they should be outside the building, at least 2 ft. below the surface and 30 ft. from any building, with the top of the tanks below the lowest pipe in the building used in connection with the apparatus. In small and medium

sized installations, steel tanks coated with tar, having a capacity of 8500 to 15,000 gal. each, or 200 to 370 bbl. of 42 gal., are generally used for storage. In larger installations, reinforced concrete tanks, generally rectangular in shape, are used. These are usually made with a partition in the center, so that any sediment or thick material may be periodically cleaned out without interfering with the continuous supply of fuel. The capacity of the storage tanks may vary from a supply sufficient for two weeks, when the oil is near at hand, and more may be obtained on one day's notice, to a supply sufficient for two or three months when the source of supply is at a considerable distance and delivery is in large quantities at irregular intervals.

17 Storage tanks should be fitted with vent pipes, indicators showing level of oil in tanks, filling pipes, arrangements for freeing tanks from water, suction pipes, return or overflow pipes, steam pipes for filling space in tanks above oil with steam in case of fire, and suitable manholes for cleaning out purposes. A suitable strainer should be installed on the suction line between the storage tanks and the oil-pressure pumps. The suction line should slope so that it will drain all oil back to the storage tanks when the pump is stopped and a vent opened.

18 Duplicate oil-pressure pumps should be installed with pump governors and all piping in connection with these pumps should be cross-connected in such a manner that a change can be made from one to the other and repairs made to either without interrupting the service.

19 A suitable oil heater should be installed, so that the exhaust steam from the oil pumps can be utilized to heat the oil before it reaches the burners. A suitable relief valve should be installed on the discharge line between the pumps and the burners set at a definite maximum oil pressure.

20 An oil meter should also be installed in the discharge line to check the storage-tank indicator readings. All oil piping should be installed so that it can be drained back to the storage tanks by gravity in case of necessity.

21 Provision should be made for removing any condensation from the steam lines to the burners. Automatic regulating devices should be installed to vary the pressure of both oil and steam to the burners in accordance with the demand for steam on the boilers, thus keeping a uniform steam pressure with a variable load, relieving the fireman of constant adjustment of burner valves and enabling him to take care of a much larger capacity of boilers than he otherwise could.

22 In case a plant is operated only ten hours per day, no steam being required for the rest of the 24 hours, it is necessary to install a small auxiliary boiler for the purpose of providing steam to atomize the oil while firing up the main boilers. In case of horizontal return tubular or Heine boilers, the burners should be placed at the front, firing towards the bridge wall. However, in the case of boilers of the Babcock and Wilcox type, higher efficiency can be obtained by placing the burners at the bridge wall and firing towards the front of the boiler.

23 It is sometimes an advantage to be able to change quickly from oil to coal and from coal to oil. This arrangement can generally be provided for, although in some cases, on account of a lack of sufficient combustion chamber space, a more efficient furnace may be installed by making the change back to coal a somewhat longer process, requiring the insertion of bearer bars and grate bars, which would be left in place in the arrangement first referred to.

TESTS OF OIL-BURNING BOILERS

24 Table 2 gives the data and results of a series of tests recently made by The Pacific Light and Power Company at their plant, Redondo, Cal., on a 604-h.p. Babcock and Wilcox boiler equipped with Hammel furnace and burners. The boiler was in regular service and under usual plant operating conditions. The ash-pit doors were wide open during all tests. The tests were under the direction of Frank T. Clarke of the power company. Six of the tests were conducted on six consecutive days. The boiler used for testing was No. 1, in battery 1, erected in 1906; heating surface, 6042 sq. ft. The oil used was crude from the Los Angeles fields, the calorific value being obtained by the New York Testing Laboratory, Los Angeles branch.

25 It will be noted that the average temperature of the flue gases is 435.5 deg. fahr. with a minimum of 385.3 deg. fahr. at 72.7 per cent of rating, and a maximum of 537.5 deg. fahr. at 195.5 per cent of rating. The average per cent of CO_2 is 13.2, and excess air 21.2 per cent; steam used by burners 2.15 per cent. The highest efficiency obtained was 83.3 per cent. This was on the test of August 11, while running at 109.2 per cent of the builder's rating, and the water evaporated per pound of oil from and at 212 deg. fahr., corrected for moisture, was 15.81 lb. The average efficiency for all seven tests, running from 72.7 per cent up to 195.5 per cent of rating, was 80.47 per cent, and the average evaporation from and at 212 deg. fahr. was 15.23 lb.

26 In Table 3 are given results of tests made at the Ravenswood plant of The New Amsterdam Gas Company, Ravenswood, N. Y. These tests were made on a 595-h.p. Babcock and Wilcox boiler, equipped with a Peabody furnace and four No. 1 burners. The tests were made for the company by N. E. Lewis. The boiler had 5946 sq. ft. of boiler heating surface, and 841 sq. ft. of superheating surface.

TABLE 2 GENERAL DATA AND RESULTS OF

Date of test, 1910	Aug. 8
Test number	1
Duration of test, hours	7
Steam pressure, lb	184.9
Steam temperature, deg. fahr	473.3
Superheat, deg. fahr	91.8
Feed water temperature, deg. fahr	90.6
Factor of evaporation	1.237
Average water level, in	4
Barometer, in	29.97
Temperature boiler room, deg. fahr	88.7
Temperature flue gases, deg. fahr	385.3
Draft, in. — In ash pit	0.025
Draft, in. — In furnace	0.01
Draft, in. — In rear pass	0.005
Carbon dioxide, per cent	12.2
Oxygen, per cent	3.6
Excess air, per cent	28.7
Smoke	None
Temperature first pass above third tube, deg. fahr	1100
Temperature top first pass, deg. fahr	640
Temperature top second pass, deg. fahr	570
Temperature bottom second pass, deg. fahr	500
Temperature bottom third pass, deg. fahr	450
Total water actually evaporated, lb	85766
Total water evaporated from and at 212 deg. fahr., lb	106092
Water evaporated from and at 212 deg. fahr., lb. per hr	15156
Steam used by burners, lb. per hr	234
Steam used by burners, per cent of total steam	1.54
Steam pressure to burners, lb	48.2
Oil pressure to burners, lb	11.4
Temperature of oil to burner line, deg. fahr	131.3
Specific gravity of oil at 60 deg. fahr	0.9770
Specific gravity of oil, Baumé, at 60 deg. fahr	13.3
Moisture in oil, per cent	0.4
Heat value of oil as fired, B.t.u	18280
Heat value of oil corrected, B.t.u	18353
Total oil as fired, lb	6913
Total oil corrected for moisture, lb	6885
Oil fired, lb. per hr	987
Oil corrected, lb. per hr	983
Water evaporated from and at 212 deg. fahr. per sq. ft. heating surface, lb	2.58
Boiler horsepower builder's rating	604
Boiler horsepower developed	439.3
Per cent of builder's rating	72.7
Water evaporated per lb. of oil from and at 212 deg. fahr. — Oil as fired, lb	15.35
Water evaporated per lb. of oil from and at 212 deg. fahr. — Corrected for moisture, lb	15.41
Boiler efficiency, per cent	81.1

The fuel used was a low grade of fuel oil, hardly higher in heat value than gas-house tar. Attention is called to the fact that only 1.43 per cent and 1.54 per cent respectively of the steam generated was used to atomize the oil in the burners. The two series of tests show the results that can be obtained with good oil-burning equipment well installed and properly operated.

TESTS ON OIL-BURNING BOILER, REDONDO, CAL.

Aug. 9	Aug. 10	Aug. 11	Aug. 12	Aug. 13	Sep. 5	Average of all tests
2	3	4	5	6	7	
7	7	7	7	7	7	7
184.9	186.0	184.7	183.5	184.6	184.9	184.8
457.4	468.8	465.1	473.8	493.8	526.7	479.8
76.0	86.9	83.7	93.0	112.5	144.3	98.3
92.6	92.7	93.4	90.8	94.6	101.2	93.7
1.225	1.232	1.229	1.237	1.245	1.259	1.238
4	4	4	4	4	4	4
30.00	30.00	30.09	30.08	30.04	29.68	29.97
86.6	84.4	85.2	87.3	86.7	84.4	86.2
397.5	409.1	406.2	429.0	477.1	537.5	434.5
0.035	0.055	0.044	0.071	0.127	0.230	0.084
0.005	0.025	0.014	0.060	0.130	0.188	0.062
0.014	0.061	0.046	0.133	0.296	0.471	0.147
13.4	13.3	14.3	14.2	13.3	12.1	13.2
2.7	2.4	1.8	1.7	2.8	6.8	3.1
17.7	18.5	10.6	11.3	18.5	43.0	21.2
None	None	None	None	None	Light haze	.....
1090	1160	1180	12.40	1300	1600	1240
640	700	680	780	940	1170	793
540	620	610	650	740	820	650
500	520	510	550	600	700	554
450	505	495	530	570	660	523
111988	129628	129609	156622	191290	226558	147351
137185	159702	159289	193741	238156	285236	182771
19598	22814	22756	27677	34022	40748	26110
441	549	549	624	708	873	568
2.25	2.40	2.40	2.25	2.08	2.13	2.15
74.7	99.7	102.6	120.4	142.5	167.6	122.2
15.2	24.4	25.4	38.3	46.1	61.6	31.6
134.3	133.5	142.3	140.1	142.1	141.7	137.9
0.9770	0.9763	0.9770	0.9776	0.9776	0.9797	0.9776
13.3	13.4	13.3	13.2	13.2	12.9	13.2
0.5	0.45	0.4	0.8	0.65	0.6	0.54
18256	18131	18253	18214	18171	17985	18184
18347	18212	18326	18357	18289	18093	18281
8758	10322	10115	12602	16580	20205	12213
8714	10276	10075	12501	16472	20084	12144
1251	1474	1445	1800	2369	2887	1745
1245	1467	1439	1786	2353	2869	1735
3.24	3.78	3.77	4.58	5.63	6.74	4.32
604	604	604	604	604	604	604
568.0	661.3	659.6	802.2	986.2	1181.0	756.8
94.0	109.4	109.2	132.8	163.3	195.5	125.3
15.66	15.47	15.75	15.37	14.37	14.12	15.15
15.74	15.54	15.81	15.49	14.46	14.20	15.23
82.8	82.4	83.3	81.5	76.4	75.8	80.47

RELATIVE COST OF OIL AND COAL

27 A tabulation could be made showing the number of barrels of oil equivalent to a ton of coal of varying calorific value, but this would

TABLE 3 GENERAL DATA AND RESULTS ON TESTS ON OIL-BURNING BOILER, RAVENSWOOD, N. Y.

Number of test		4760	4761
Date of test, 1907		July 2	July 3
Duration of test, hours		5	4
Steam for atomizing, superheated or saturated		Saturated	Saturated
Steam pressure by gage, lb		145	142
Temperature of feed water, deg. fahr		132.4	119.6
Degrees of superheat, deg. fahr		85	83
Factor of evaporation		1.1750	1.1868
Total oil burned, lb		8273	6687
Oil burned per hour, lb		1655	1672
Total water evaporated, lb		101546	82317
Total water evaporated from and at 212 deg. fahr., lb		119317	97694
Water evaporated per hour from and at 212 deg. fahr., lb		23863	24423
Draft, boiler side of damper, in		0.29	0.29
Draft in furnace, in		0.16	0.15
Temperature waste gases in flue, deg. fahr		488	488
Temperature air in boiler room, deg. fahr		89	82
Temperature oil at burners, deg. fahr		97	97
Flue gas analyses	Carbon dioxide, per cent	13.14	13.25
	Oxygen, per cent	5.29	5.15
	Carbon monoxide, per cent	0	0
Water evaporated per sq. ft. boiler heating surface per hour from and at 212 deg. fahr., lb		4.01	4.11
Water evaporated per sq. ft. total heating surface per hour from and at 212 deg. fahr., lb		3.52	3.60
Water evaporated from and at 212 deg. fahr. per lb. of oil		14.42	14.61
Horsepower developed		691.7	707.9
Per cent of rated capacity developed		116.2	118.9
Steam used by burners per hour, lb		340	377
Per cent of steam generated used by burners		1.43	1.54
Net water evaporated from and at 212 deg. fahr. per lb. of oil (allowing for steam used by burners)		14.21	14.38
B.t.u. per lb. of oil		17733	17425
Efficiency based on gross evaporation		78.53	80.97

be of very little practical value taken alone, for the reason that the efficiency of a boiler using oil fuel would be from 5 per cent to 15 per cent higher than when using coal, due to more perfect combustion with oil. The gain in efficiency would vary with the size of the boilers, character of furnaces and other equipment, intelligence of firemen, etc.

28 Although a fair idea may be obtained of the comparative cost of the two fuels by making certain assumptions in regard to heat

values, specific gravity, gain in efficiency, etc., still this will not enable one to figure the saving which could be made by changing from one fuel to the other. The reason for this is that the saving generally depends on other things than the cost of the fuel. The saving in firemen and coal passers, increase in capacity, facilities for fuel storage, advantage of pumping fuel over methods of handling coal, elimination of handling ashes, quantity of coal used for banking fires, elimination of smoke and other things, many of which cannot be figured out in advance in dollars and cents, would throw the ultimate cost decidedly in favor of oil. The only way to determine the exact saving is to operate the plant with each fuel for a long enough period to get accurate data on all the items entering into the question.

29 The writer also believes that the success of any oil-fuel installation depends less on the type of burner or atomizer used than on the general efficiency of the entire installation and the intelligence with which it is operated after the installation is made. Therefore, the work of designing and constructing such installations should be entrusted only to those having had extended practical experience in burning oil fuel and the operation of these installations should not be given to unskilled labor, for, in order to obtain the economic results possible from the use of oil fuel, it will be necessary to employ men of intelligence and skill as firemen.

APPENDIX No. 1

30 A few typical burners representative of the subdivisions and classes mentioned in Par. 11 are shown in Figs. 1-13. Although these cover only a few of the burners and furnaces now in use in this country, not to mention the large number of systems of oil burning, the writer believes that they are sufficiently typical to show the general principles involved, which is all he can attempt in this paper. The author is responsible for the general classification given in the case of each burner, but the details of construction and operation are as given by the manufacturers in their descriptive literature:

Fig. 1 Peabody No. 1. Outside mixing. Drooling. Fan-shaped flame. Spraying from sharp edge. The cut shows clearly the details of construction, including oil strainer, steam by-pass for blowing out oil passages, oil pipe jacketed by steam and removable burner tip.

31 This tip contains two very narrow slots separated by a diaphragm, the lower slot for steam and the upper for oil. The oil falls at right angles upon the steam jet which atomizes it and the oil is burned in a fan-shaped flame. It will be noted that the tip is held in position by a single bolt so that it can be readily cleaned if necessary. The mixing or atomizing is done entirely outside the burner. This style is for use in connection with the Peabody furnace, in which the burner tip is located at the bridge wall and the flame projected towards the front of the boiler.

Fig. 2 Peabody No. 2. Similar to Peabody No. 1 except that it is designed to be placed at the front of the boiler and to project the flame toward the bridge wall in the ordinary manner.

Fig. 3 W. N. Best high-pressure oil burner. Outside mixing. Atomizing. Long slot. Spraying from sharp edge. The air or steam meets the oil at right angles, thus atomizing it externally. Either a long narrow flame or a fan-shaped flame can be provided as required.

Fig. 4 W. N. Best oil burner with piping connections. Same as Fig. 3, except fitted with piping of sufficient length to go through the front setting of boiler. By-pass is provided for blowing out any foreign substance that enters the oil pipes. The atomizer lip is hinged and means provided for raising the lip for blowing-out purposes without removing the burner from the boiler.

Fig. 5 Gem oil burner. Outside mixing. Drooling. Rose-shaped orifice. Spraying aided by centrifugal action from internal screw and cone. Oil heated by atomizing steam on the way to burner tip as it surrounds oil pipe.

Fig. 6 Gilbert and Barker oil burner. Outside mixing. Drooling Rose-shaped orifice. Spraying aided by slight centrifugal action from internal helix. Adapted for use where very heavy consumption of oil is required, such as heavy metallurgical operations, brick kilns, etc.

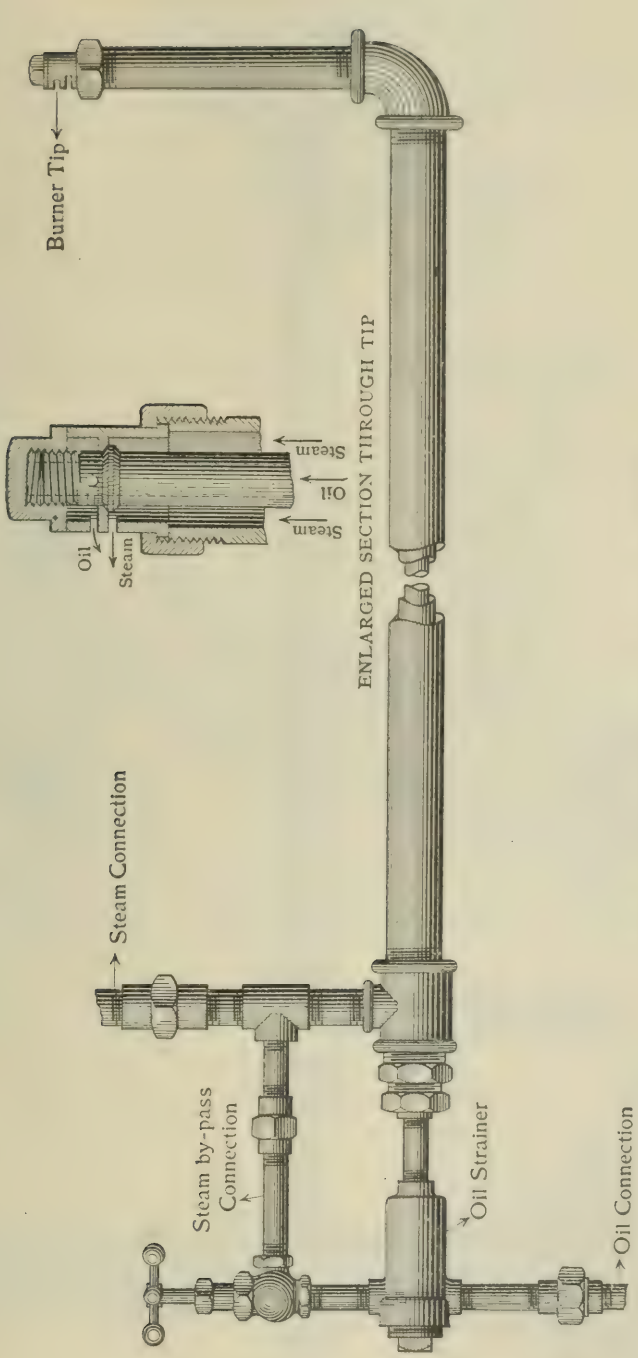


FIG. 1 PEABODY NO. 1 OIL BURNER

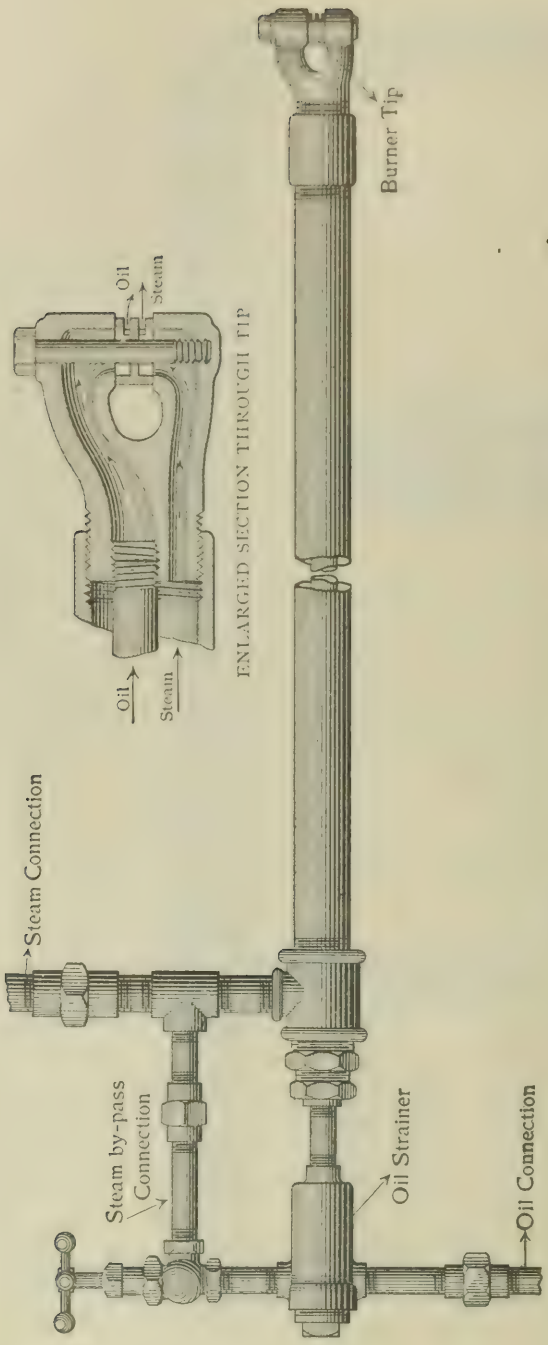


FIG. 2 PEABODY NO. 2 OIL BURNER

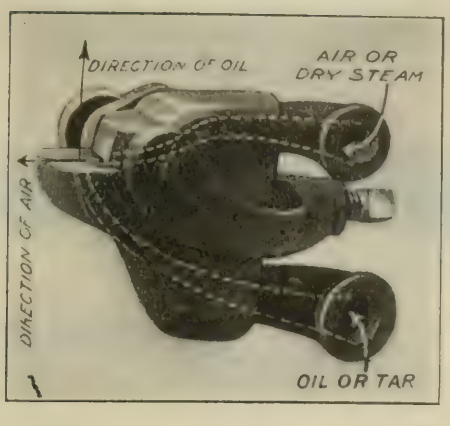


FIG. 3 W. N. BEST HIGH-PRESSURE OIL BURNER

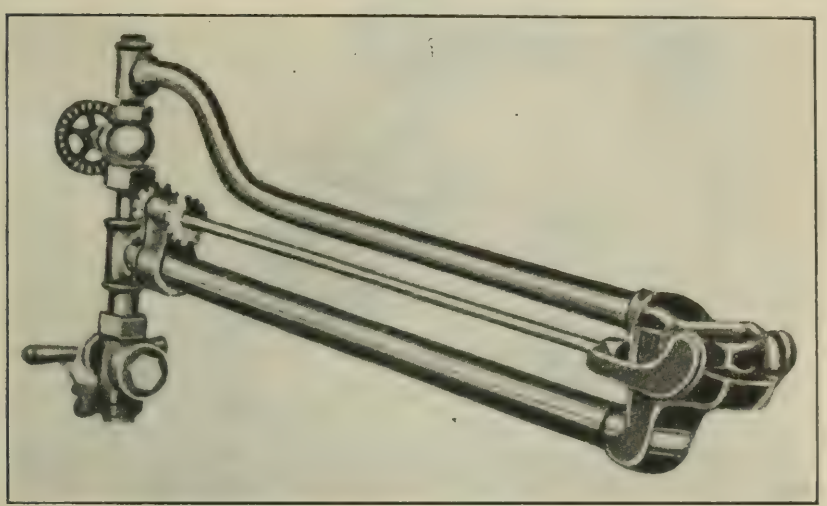


FIG. 4 W. N. BEST HIGH-PRESSURE OIL BURNER WITH PIPING CONNECTION

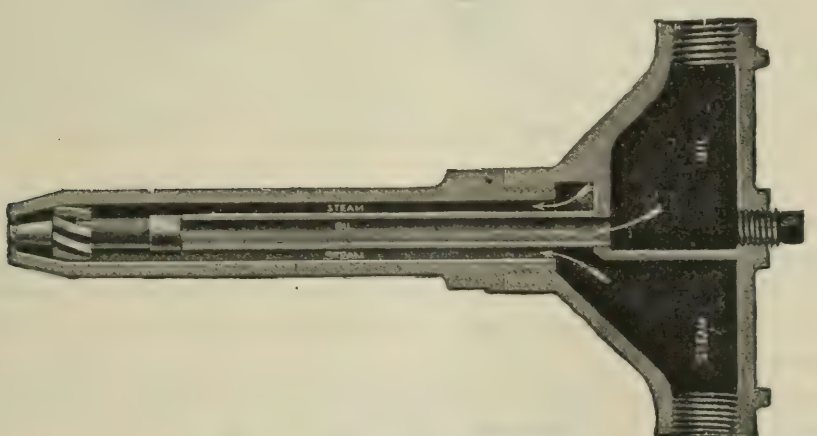


FIG. 5 GEM OIL BURNER

Fig. 7 Rockwell high-pressure oil burner. Outside mixing. Atomizing. Steam or air and oil nozzles removable for cleaning-out purposes. Steam or air nozzle outside of oil nozzle. Adapted for light or heavy work.

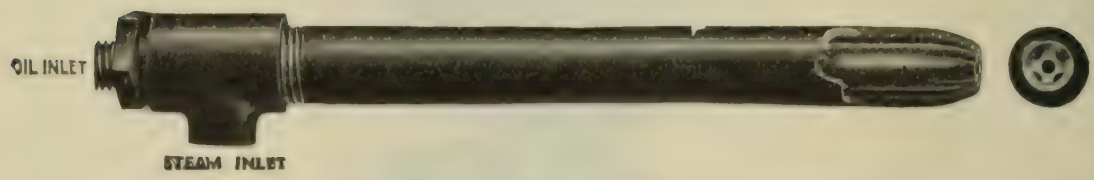


FIG. 6 GILBERT AND BARKER OIL BURNER

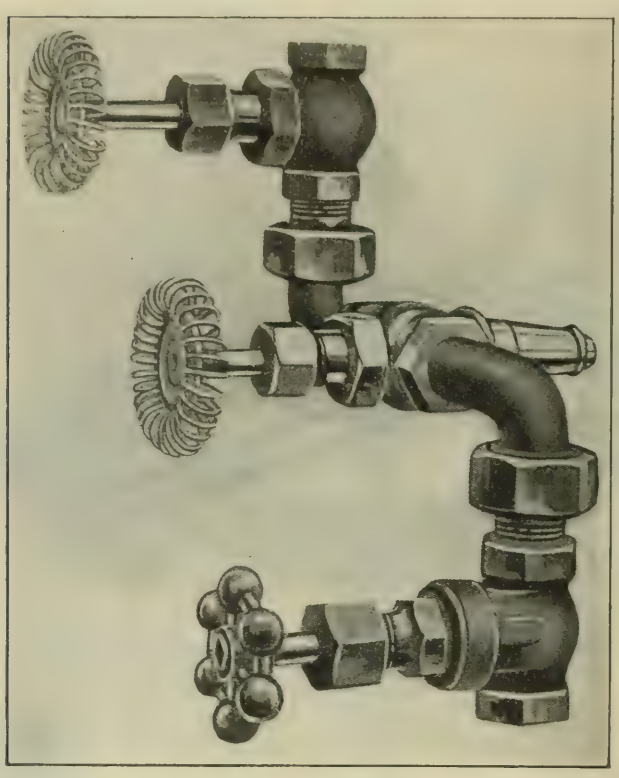


FIG. 7 ROCKWELL HIGH-PRESSURE OIL BURNER

Fig. 8 Hammel oil burner. Inside mixing. Chamber and atomizing. Long slot. Spraying from sharp edge. All parts are named in the cuts. Oil enters at *A* and flows through *D* into mixing and atomizing chamber *C*; steam enters at *B* passes through *F*, *E* and then through three small slots, *G*, *H* and *I* into mixing chamber *C*, where it meets the oil. The small steam jets break up the oil in the mixing chamber and the mixture is ready for ignition as it issues from the fan-shaped orifice. This orifice is provided with removable steel plates, which can be replaced easily in case of wear.

Fig. 9 Kirkwood oil burner. Inside mixing. Injector. Rose-shaped orifice. Spraying from sharp edge. Both oil and steam valves seat at tip of the burner and are equipped with indexes showing the amount of oil and steam turned on.

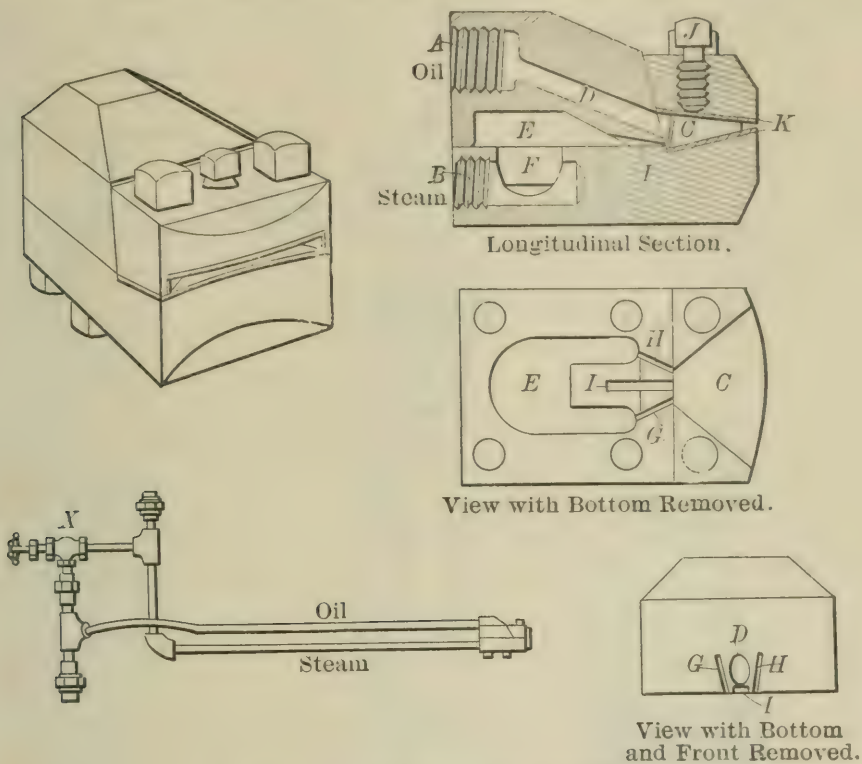


FIG. 8 HAMMEL OIL BURNER

A Orifice for Oil Supply Pipe. *B* Orifice for Steam Supply Pipe. *C* Mixing or Atomizing Chamber. *D* Oil Inlet Duct. *E* Equalizing Steam Chamber. *F* Steam Entrance. *G*, *H*, *I* Steam Ducts. *J* Set Screw holding Plate. *K* Removable Steel Plates. *X* By-Pass or Blow-Out Valve.

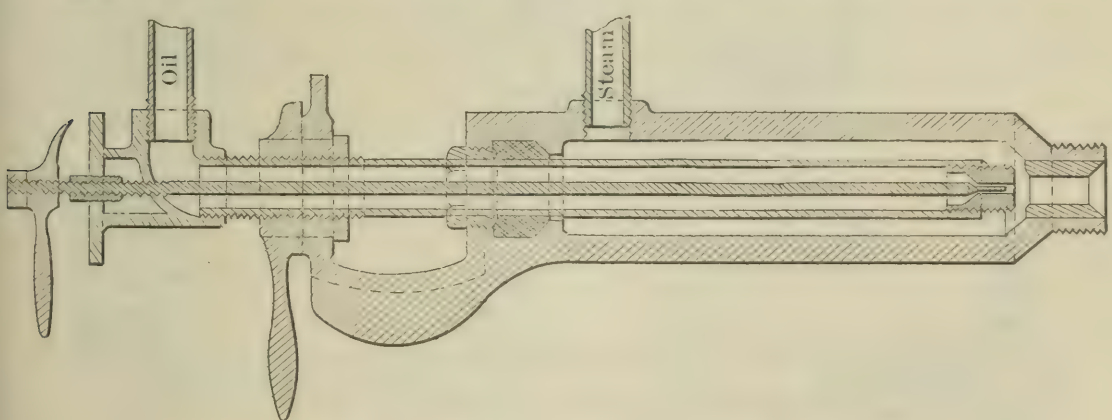


FIG. 9 KIRKWOOD OIL BURNER

In another style of this burner one lever controls both the oil supply and the steam necessary for atomizing, the proportion being fixed before shipment from factory.

Fig. 10 Improved Little Giant oil burner. Inside mixing. Chamber. Double long slot. Spraying from sharp edges. Oil pipe surrounded by steam

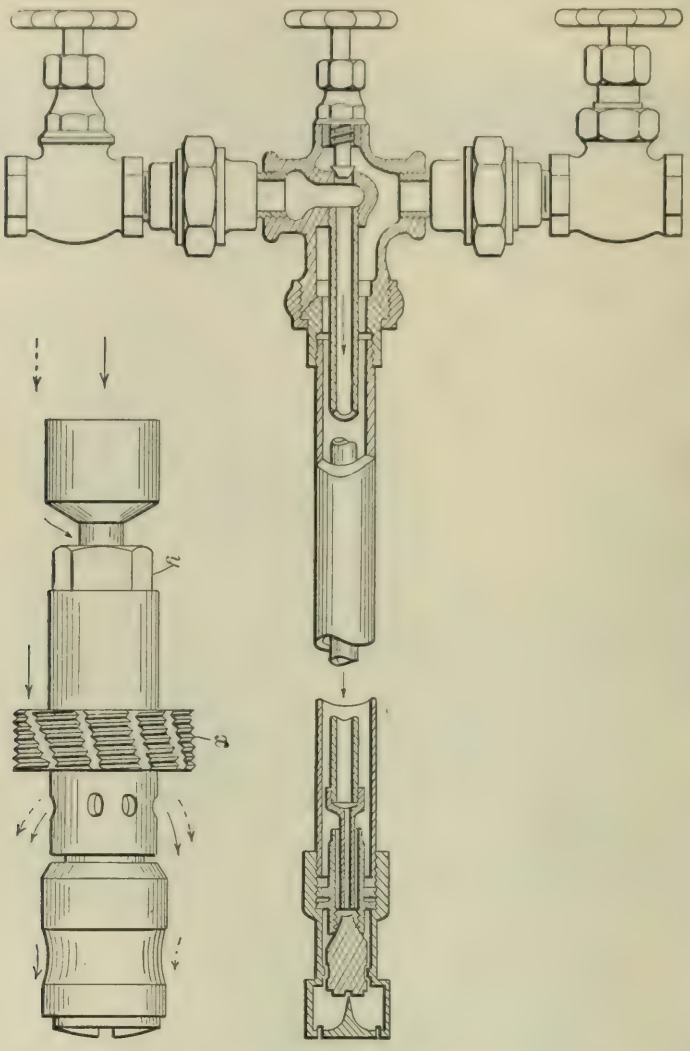


FIG. 10 IMPROVED LITTLE GIANT OIL BURNER

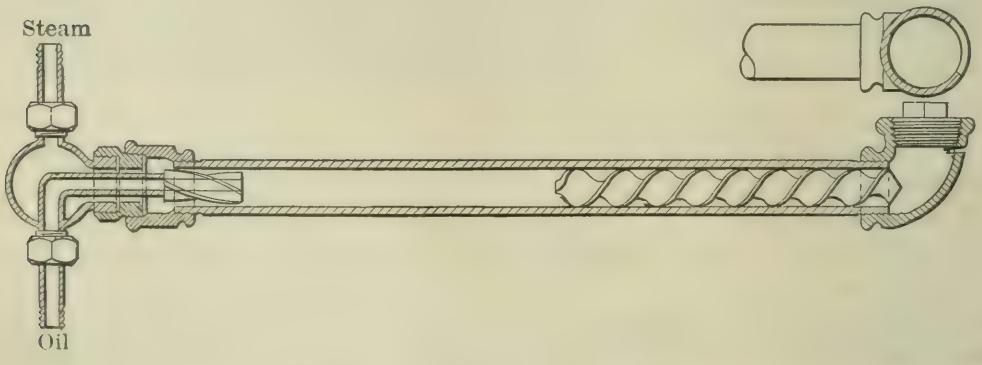


FIG. 11 TEXAS OIL BURNER

which enters mixing chamber near tip. A separating diaphragm causes an equal amount of the mixture of oil and steam to issue from each opening in the tip. A special non-clogging oil valve is used.

Fig. 11 Texas oil burner. Inside mixing. Chamber. Fan-shaped flame. Spraying aided by centrifugal action from internal screw. As the oil flows into

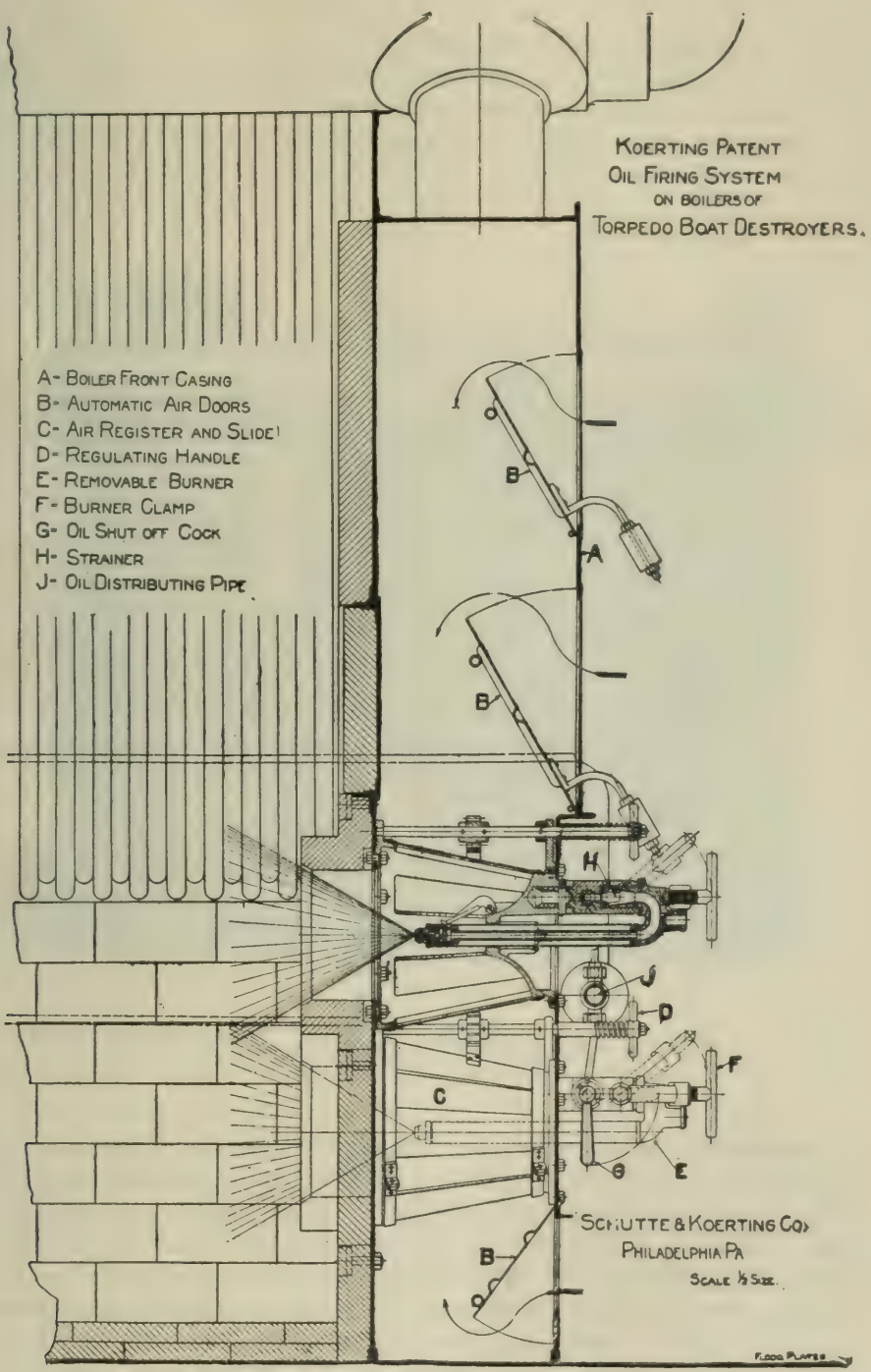


FIG. 12 KOERTING OIL BURNER

the large mixing chamber it is picked up by the steam to which rotary motion has been imparted by a short helix in the steam passage just back of the oil inlet. The mixture then passes along the chamber through a spiral passage occupying about one-half of its length which sets up a strong centrifugal action so that the oil is thoroughly atomized and vaporized when it issues from the fan-shaped orifice in the small chamber at the tip of the burner. This orifice is made to give any width of flame required and the tip is easily renewable in case of wear.

Fig. 12 Koerting oil burner. Mechanical spraying by pressure and centrifugal action from internal screw. Oil is delivered to the burner under sufficient pressure to break it up into a conical spray with the aid of spiral blades which set up a strong centrifugal action and a central cone at the tip which helps to maintain the centrifugal effect until the oil issues from the orifice. By loosen-

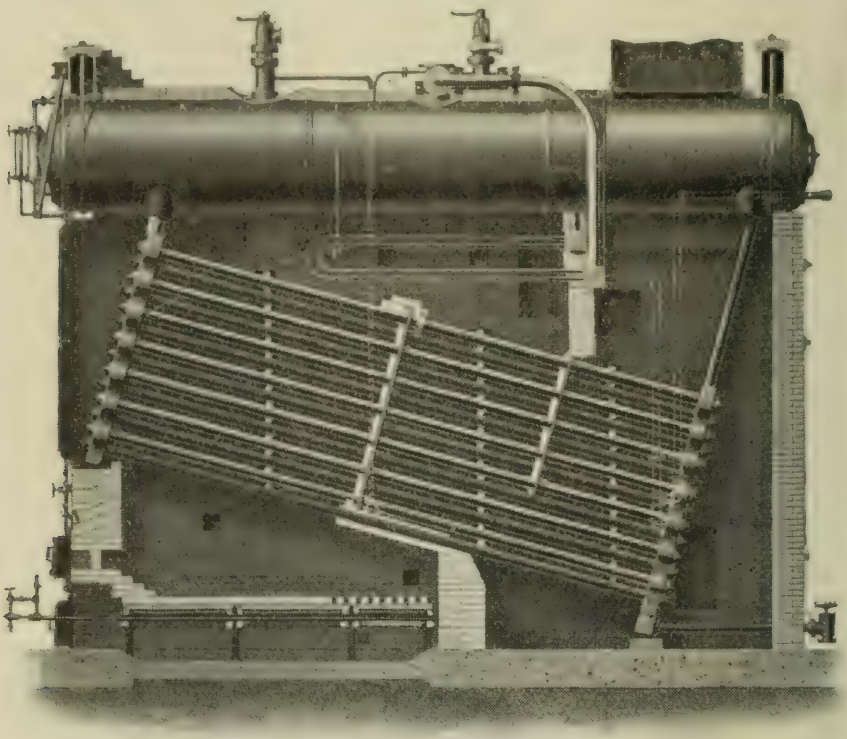


FIG. 13 PEABODY FURNACE FOR LIQUID FUEL

ing the clamp screw and throwing it into the dotted position shown, all working parts of the burner are quickly removable for examination or cleaning. The supply of air is introduced through adjustable openings shown in the cut so that it will impinge directly on the spray of oil thus effecting immediate combustion. The fact that this burner requires neither steam nor compressed air for atomizing makes it especially desirable for marine and naval boilers where fresh water for boiler feed make-up purposes is expensive.

Fig. 13 Peabody furnace for liquid fuel. E. H. Peabody of the Babcock and Wilcox Company, while making an exhaustive study several years ago of

methods of burning oil under water-tube boilers, discovered that by placing the burners at the bridge wall and projecting the flame toward the boiler front a large gain in efficiency and capacity could be made. This gain is due to the thorough distribution of the flame through the furnace and the complete combustion of the gases before coming into contact with the boiler tubes as well as the consequent improved utilization of the entire heating surface in the first pass of the boiler. All details of the furnace are clearly shown in the cut, including extension of furnace towards the rear of the boiler, arrangement of air openings, method of supporting burners and inserting them through special slide ways supported on the bearing bars which hold the floor of the furnace.

32 The Hammel furnace, used in the tests in Table 2, is based on the Peabody patent furnace (Fig. 13) with the addition of a slot in the bridge wall in which the burner is placed for protection from the heat and a separate air tunnel to each burner so that one or more burners may be extinguished and their air supply entirely shut off, thus maintaining the efficiency of the burners still in use.

DISCUSSION

E. H. PEABODY judged that the reference to the steam consumption of the oil burners, in Par. 24, pertained to the tests made by N. E. Lewis at the Ravenswood works of the New Amsterdam Gas Company, the results of which are given in Table 3. These tests were two incidental runs made in a series of experimental tests mainly with gas-house tar as fuel. It was not expected that they would achieve the distinction of appearing in the proceedings of the Society, and while very good tests, they should not be taken as a fair comparison of results obtained with Hammel burners, as their proximity to Table 2 of the paper would seem to indicate.

M. H. BRONSDON[1] stated that every one of the tests of crude or fuel oil made for him by Professor O'Neill at the University of California showed the same calorific value (practically), regardless of its specific gravity or whether or not the gasolene had been removed. The more fluid the oils, the less troublesome they were, as they required a much lower pressure to send them through the piping; and where the specific gravity is 17 deg. Baumé, or lighter, at 60 deg. fahr., no warming during the pumping process was necessary.

The use of fuel oil removes one of the most important and expensive items of power plant operation from the hands of unskilled labor (i.e., the ordinary fireman), and places it upon an efficient basis, as with proper installation economical operation is dependent only upon reliable and very simple mechanism.

[1] Chief Engineer, The Rhode Island Co., Providence, R. I.

Where oil is used for fuel, perfect combustion may be obtained under all conditions of load with proper installation, excepting when the fires are first lighted and the brick work is comparatively cold. Boiler settings and boiler tubes last much longer with fuel oil than with coal, provided the tubes are kept clean.

The impression should not be given, as in Par. 2, that by changing from coal to fuel oil the capacity of any boiler plant can be increased to from 35 to 50 per cent. In boiler plants with sufficient draft to burn large quantities of coal, or where the evaporation can be made to exceed, say 7 lb. of water per sq. ft. of heating surface, fuel oil will not increase the capacity 35 to 50 per cent.

His experience led him to favor burners of the type known as "outside mixers," i.e., where the oil and steam mix just beyond the tip of the burner. Carbon seldom causes trouble with burners of this type, even where the oil is quite hot before it reaches the burner. There seems to be no practical difference in the efficiency, however, in the use of either inside or outside mixer burners.

Comparing the cost of fuel oil with coal, we should consider, first, the cost of coal alongside, plus discharging costs, plus conveyor costs, plus the cost of removing ashes; second, the B.t.u. per ton of coal. In New England at tidewater, the cost of coal, and the discharging and conveying costs and the removal of ashes approximate $3.50 per ton, varying from year to year. One ton of bituminous coal contains approximately 33,000,000 B.t.u. and we should buy 33,000,000 B.t.u. in fuel oil for approximately $3.50; i.e., 33,000,000 B.t.u. equals

1774 lb	$3.50
224 gal	3.50
1 gal	0.0156
1 bbl	0.65

These figures will be slightly reduced if credit is given for minor economies resulting from the use of oil, but they are amply accurate for general use.

D. S. Jacobus said that the efficiency results given in Table 2, secured in tests on an oil-burning boiler at Redondo, Cal., represented good practice. As good and slightly better results, were secured in tests on one of the boilers at the same plant preparatory to making a test of the plant. The plant tests, already reported to the Society, indicated that a kilowatt-hour was turned out at the switchboard for each 25,000 B.t.u. contained in the fuel oil. In these tests the

standard form of Peabody furnace was employed with burners of the outside mixer type.

S. F. McIntosh[1] believed that there was little choice between the two when figuring on oil at $0.03 a gallon and coal at $5 a ton delivered in the boiler house on the B.t.u. basis. At the plant of the American Optical Company, Diesel engines were installed, and as more or less oil equipment was required for their operation, it was decided to try oil under the boilers. It has worked almost perfectly and there has been no trouble whatsoever. Its advantages in the way of ease of operation, cleanliness, efficient combustion, etc., appeal to them strongly. It should be noted, however, that a slight increase in the price of oil would offset the saving of labor through its use. Taking the statement of the author that one man can handle four 500-h.p. boilers under which oil is used, where four men would be required if coal were used, and assuming the saving in labor to be 80 cents an hour, an increase of $\frac{1}{10}$ cent in the price of oil would entirely offset the saving in labor.

B. R. T. Collins referred to the fact that Mr. Bronsdon had had his experience with fuel oil in California, where the crude oil is quite different from the residue from Texas, which would alter his conclusions somewhat. The reason for more complete combustion referred to by Mr. McIntosh is that there is no loss in burning oil such as occurs in burning coal, at the period just after the coal is fired, especially with hand firing. At that time a large quantity of hydrocarbons are only partly consumed. It therefore stands to reason that an oil-burning equipment should be more efficient. There are more heat units of the oil utilized in producing steam than of the coal in an ordinary coal-burning installation.

C. F. Dietz said that the two methods used for atomizing oil call either for steam or air and it would appear that the steam blast might have advantages over the air blast in the way of flame temperature and the consequent effect upon the surfaces where the flame impinges. He was not aware of the existence of any data regarding the effect of the dissociation of steam on the temperature of the flame, nor did he know whether the reverse action would take place almost immediately so that the heat taken up by the dissociation of the

[1] American Optical Co., Southbridge, Mass.

water would be partly returned to the boiler as useful energy. For the purpose of reviewing the chemistry and physics of the subject, he presented calculations indicating to him that the steam after dissociation must necessarily be burned back to water at a point in the boiler where the heat thus produced from the exothermic reaction is realized.[1]

J. C. Riley called attention to the fact that the flame from an oil burner is much like that from a blast lamp; its very high temperature is localized over so small an area that the metal of the boiler must be protected from it, whether the atomization is by steam or air. He had measured some temperatures in the furnaces of Scotch boilers with burners fired by both air and steam atomization. The furnaces were of the usual corrugated cylindrical type, opening into separate combustion chambers at the back. Each had a complete ring of fire-brick lining which extended about 4 ft. back, and another ring in front of and around the saddle seams where they entered the combustion chambers. There was also a layer of fire brick along the bottom, between the two rings, and one burner in each furnace, placed just above the axis and terminating about a foot back of the tube sheet.

On looking in at the front, the burner could be seen to spread the flame in a cone of about 80 deg. total angle, striking on the ring of brick. Apparently it was hottest right where it struck. With the steam burners, the flame was so dazzling that it was quite impossible to see beyond the surface of the cone. With the air burners clean and properly adjusted, the flame was yellower and not so bright; a little practice enabled one to see through the cone, even into the combustion chamber, and it appeared that the whole furnace was filled with a long, transparent yellow flame. With the air burners partially clogged with carbon, or not properly adjusted for air supply, the flame was irregular and smoky; drops of oil could be seen spattering on the furnace walls.

It was found that with an air burner just cleaned and in best condition, the temperature throughout the furnace, from the burner back into the combustion chamber, was remarkably uniform, and that it averaged about 2250 deg. fahr. It was hottest at the axis of the furnace, a foot or two in front of the burner, and not where the cone of flame struck the brick lining. After several hours use without cleaning, the air burner gradually clogged with carbon until

[1]The Journal, August 1911, pp. 959-961.

the mean temperature dropped to less than 1500 deg., in which case the front of the furnace was comparatively cool and the highest temperature was in the combustion chamber. The steam burner never clogged with carbon, but its temperature was not so uniformly distributed as that for air. In the axis of the furnace it dropped from more than 2400, just beyond the burner, to less than 2000 in the combustion chamber. It was extremely hot where the cone of flame struck the brick along the bottom; one observation gave more than 2500 deg., a temperature which destroyed a quartz tube of the pyrometer.

The necessity for protecting the metal of a boiler from direct contact with the hottest point of flame is apparent to any one who has seen the flame marks which are sometimes burned in by careless firing. Burning of the steel is most likely to occur when the oil fuel contains much sulphur. It is also likely to occur when for any reason the rate of heat conduction is slow or the circulation is not active.

For burning oil it is especially desirable that the fireman should be a man of intelligence and good judgment. A few minutes of neglect or carelessness may injure the boiler seriously. When everything is operating well and the burners properly adjusted, a slug of water may come along with the fuel and put the flame out. To be sure it ought not be there, for the suction pipe of the fuel pump is floated near the surface of the oil in the supply tank, to guard particularly against this happening. But sometimes the thing which cannot happen does occur, and the flame goes out. Then when the next oil passes the burner, if there are other burners still lighted in the same furnace, or if the brick target is still hot enough to ignite the oil instantly upon contact, no harm is done; but if the oil does not light, and if the fireman does not notice the fact, there is great danger of an explosion, or at least a fire.

No. 1309

THE ECONOMIC IMPORTANCE OF THE FARM TRACTOR

By L. W. Ellis,[1] La Porte, Ind.

Non-Member

The lack of power for plowing and harvesting is the tremendous obstacle to the sudden expansion into virgin fields of our productive area. The shallow plowing now generally practised consumes 60 per cent of the total power expended in raising and harvesting the wheat crop, even on old land. Deeper plowing to secure maximum yields sharpens the necessity for power in the brief plowing season. The slow process of animal reproduction cannot respond quickly enough, and the price of horses has increased 143 per cent in ten years in spite of a 50 per cent increase in the supply. Today in Canada, where great added power is imperative, horses can be purchased only in limited numbers. Even the United States Department of Agriculture cannot find an adequate supply of brood mares for the future needs of the New South. Increased production cannot safely depend on animal power.

2 Nor is production the only consideration. Fifteen million work animals, and the 10,000,000 more to keep up the supply, scarcely develop sufficient power for present farm purposes. Their feed alone costs $1,250,000,000 per year, equalling the total income of 2,000,000 average families. Thus the crops from one acre in five are withheld from supplying human needs by the use of animals for farm power.

3 The farm tractor is the solution of the immediate problem. It does not age nor deteriorate when idle and requires neither fuel nor attendance when not at work. The time spent annually in caring for a horse will keep the tractor in perfect working condition. It

[1] Traction Plowing Specialist, M. Rumely Co.

Published in abstract form. Presented at the New York Meeting, April 1911, of The American Society of Mechanical Engineers. The complete paper appeared in The Journal for March 1910.

will endure heavy work 24 hours a day instead of 6 and outlive the average work animal in hours of service. It occupies less floor space than two wagons, and with a year's fuel supply may be sheltered in a building a tenth the size and cost required to house and maintain horses of equal power. Efficient farm labor grows increasingly scarce. The tractor concentrates in one man's hands the power of 25 horses and the endurance of 100, and adds two-fold to the acres he can cultivate.

4 As regards power, at points distant from the refineries and close to the coal fields steam engines may have the advantage of the internal-combustion type. In the nearer West the latter have come into use with a swiftness that is amazing when one considers that the first one to prove successful was launched only about eight years ago. Their economy in small sizes, their convenience, safety around buildings, and the fact that operators are not usually required to hold licenses, commend them strongly. Wonderful tank wagon service now brings fuel for the oil-burning engine direct to the farm at wholesale prices, an engine often yielding as great a profit to the refiner as a town of 600 inhabitants.

5 To date, no kerosene-burning tractor has achieved the thermal efficiency secured by the foremost gasolene tractors, but in all except the most remote districts the wide and growing disparity in fuel cost gives the former a marked commercial advantage.

6 The data which I have on the performance of tractors are largely from two sources, the agricultural motor competitions which have been held at Winnipeg and Brandon, Manitoba, and investigations which I conducted among every-day operators in connection with the United States Department of Agriculture. The former should be thoroughly reliable, but in some cases are open to doubt, the tests having been conducted hastily and without adequate facilities. The latter source is reliable only in that it averages the testimony of a large number of operators, few of whom had exact records. The tests conducted in the four motor competitions have been on the brake, first for economy, then for maximum power; in plowing, over firm, level prairie sod; and in hauling, over a half-mile circuit which presented almost every possible road condition from block pavement to loose gravel and mud. The various averages hereinafter presented are offered merely as a basis for rough comparison with the performance of motors used in other service.

COAL AND WATER CONSUMPTION

7 Table 1 shows the average coal and water consumption in the economy brake tests of steam tractors in Canada. It is not to be supposed that the load in every case was exactly at or even near the point of greatest economy, though this condition was usually aimed at. There was but one test of a compound engine on the brake, against 6 single and 11 double cylinder engines. In plowing there were 6 single and 6 double; and in hauling 1 single and 3 double.

TABLE 1 LB. OF COAL AND WATER USED PER DELIVERED H.P. PER HR.

TEST	SINGLE CYLINDER		DOUBLE CYLINDER		COMPOUND	
	Coal	Water	Coal	Water	Coal	Water
Brake.....	3.72	30.11	4.43	32.43	4.96	34.94
Plowing.....	7.46	52.8	8.87	69.0	....	.....
Hauling.....	14.18	114.6	12.84	91.1	....	.....

8 Steam tractors as a rule use from 7 lb. to 8 lb. of water per lb. of coal. Reports from 333 plowing engines of all types in the United States and Canada indicate an average of 7.67 lb. Twenty-four public brake tests show a mean of 7.78 lb.; 16 plowing tests 7.08 lb.; and 4 hauling tests 7.4 lb.; or a mean of 7.42 lb. for the 44 tests. Single cylinder engines show a range of from 5.78 lb. to 9.97 lb.; and double cylinder from 3.3 to 10.3 according to the official reports. Conditions were such however, as to arouse doubts as to the accuracy of such extreme figures. By making enough assumptions we can compare these data with those furnished by operators of 11 oil-burning steam engines in California. These men report the use of 9.4 gal. of water per gal. of oil. Assuming the oil to be of 20 deg. Baumé and to contain 20,000 B.t.u. per lb. they use 1990 B.t.u. in evaporating 1 lb. of water. The ordinary run of coal used contains not over 13,000 B.t.u. per lb., hence 1700 to 1740 B.t.u. would be furnished per 1 lb. of water.

GASOLENE CONSUMPTION

9 Table 2 gives an average of gasolene consumption in all public economy tests to date.

10 The average consumption for 27 brake tests is 0.747 lb., or a trifle over a pint per h.p-hr., the gasolene used being of 70 specific, 64 Baumé gravity. Nineteen plowing tests average 1.67 lb., and 12 hauling tests 1.91 lb. per drawbar h.p-hr. These averages are not

comparable, but the matter of tractive efficiency will be discussed later.

11 The amount of water required per brake horsepower-hour by the internal-combustion motors depends largely on the cooling system. In these tests it ranged from 0 to 2.55 lb. per h.p-hr., averaging in the neighborhood of $1\frac{1}{3}$ pints. A kerosene tractor, using water in the cylinders, consumed less than a pint of water per h.p-hr. The consumption by the evaporative type of cooler would in

TABLE 2 GASOLENE CONSUMED PER DELIVERED H.P. PER HR.

TEST	1 CYLINDER		2 CYLINDERS		3 AND 4 CYLINDERS	
	No. Tests	Lb. Fuel	No. Tests	Lb. Fuel	No. Tests	Lb. Fuel
Brake.....	11	0.567	4	0.836	12	0.965
Plowing.....	6	1.273	4	2.076	0	1.778
Hauling.....	5	1.536	1	3.97	6	1.88

several instances have necessitated replenishment of the water supply after from two to three hours of heavy work.

THERMAL AND TRACTIVE EFFICIENCY

12 No analyses have been made of fuels used at the motor contests, hence the thermal efficiency of the various engines is in doubt. It probably ranges from 4 to 6 per cent for the steam tractors, and from 6 to 25 per cent for the internal-combustion type. Neither have tests been made of the boiler, mechanical or real tractive efficiency.

13 The fuel consumption in the brake tests bears a direct relation to the number of cylinders. In the plowing and hauling tests the efficiency of the traction parts is brought also into play. Tests bringing out comparisons of the traction mechanism only have never been conducted, hence we are forced to use a crude comparison of fuel consumed and horsepower developed in the various tests. This topic deserves extended discussion, and Table 3 will serve to bring out, though only in a rough way, the influence of various factors on tractive efficiency.

14 The small, single-cylinder motors had wheels much larger in proportion to total weight than the other classes, but more weight, also, per brake horsepower developed in the economy test. Comparing only the brake and drawbar horsepower they appear to have greater tractive efficiency than the larger and heavier machines. From the comparative fuel consumption however, we may assume

that this was due rather to harder work of the engine, i.e., more brake horsepower was being developed during the plowing than during the economy brake tests. As previously stated, the road conditions were severe, every class developing in the hauling tests a lower percentage of the brake horsepower at the drawbar, and using more fuel per tractive horsepower-hour than in the plowing tests.

15 The light high-wheeled tractors with four-cylinder high-speed engines seemed to be much less affected by the adverse conditions, developing in either plowing or hauling about half as much tractive horsepower as brake horsepower, with approximately twice the fuel consumption per unit. The steam engines, with only a trifle more weight per brake horsepower than the four-cylinder gasolene trac-

TABLE 3 TRACTIVE EFFICIENCY TESTS

DETERMINED BY COMPARING FUEL CONSUMED AND HORSEPOWER DEVELOPED AT WINNIPEG, MAN., JULY 1909

Type of Engine	Total Weight per 1 in. width of Driver	Weight per Economy B.h.p.	Drawbar h.p. / B.h.p.		Fuel per B.h.p-hr. / Fuel per drawbar h.p-hr.	
			Plowing	Hauling	Plowing	Hauling
Single-Cylinder, Gasolene, Low Wheel.....	299	535	0.614	0.508	0.441	0.355
Four-Cylinder, Gasolene, High Wheel.....	407	416	0.531	0.499	0.538	0.533
Steam.....	585	456	0.566	0.293	0.499	0.344

NOTE.—This table includes four single-cylinder gasolene engines, three 4-cylinder, and four steam engines.

tors, and about the same height of drivers, but over 40 per cent more weight per inch in width of drivers, were able to turn this weight to good account in plowing on firm footing. Over the hauling course power dropped and fuel consumption rose in nearly the same proportion.

16 The drawbar pull of the gasolene tractors averaged about 17 per cent of the total weight in the hauling test and about 24 per cent in plowing. One horsepower was developed for 922 lb. of total weight. The drawbar pull of the steam engines was approximately 11 per cent of the total weight in hauling and 22 per cent in plowing. Averaging the two tests, the steam tractors have credit for 1 tractive h.p. for each 1033 lb. of weight.

17 This year the steam tractors delivered in plowing from 50 to

77 per cent of their economy load on the brake and from 37 to 58 per cent of their maximum. The dynamometer showed a mean resistance of 26 per cent of the total weight and 36 per cent of the weight on the drivers. The latter weight is probably only approximate. A mean of percentages indicates that 73.5 per cent of the weight is borne by the drivers.

18 Seven internal-combustion tractors had a mean resistance of 25 per cent of the total weight and 35 per cent of the weight on the drivers, the latter carrying 70 per cent of the total. Data on a motor truck which overcame a resistance of 33 per cent of the total weight and 79 per cent of the weight on the drivers are excluded. This motor carries only 42 per cent of its weight on the drivers, and in the plowing test was loaded with human ballast to increase the percentage.

19 The rating of tractors is far from uniform, but steam ratings are more conservative than for internal-combustion engines. The brake rating of steam tractors is seldom less than three times the tractive or nominal rating. Some confusion exists in the use of the terms nominal and tractive. In the Winnipeg motor contests the officials regarded nominal, drawbar and tractive ratings as synonymous, and the terms are used here in that sense. In tests this year every steam tractor not only exceeded both brake and tractive ratings, but developed a greater percentage of its maximum brake horsepower at the drawbar than was indicated by the ratio of tractive to brake rating.

20 Only two internal-combustion engines equalled their brake rating on the maximum test and only one its drawbar rating in plowing. The class averaged only 86.5 per cent of the brake rating on a maximum test, 80.9 per cent of the tractive rating on a semi-economy test in plowing and 80.9 per cent of their brake rating on an economy load. The steam engines exceeded the specified horsepower by 32 per cent, and the tractive rating 95 per cent, besides carrying 97 per cent of the rated brake horsepower on the economy load.

21 The daily capacity of a tractor depends on the width of the strip plowed or otherwise treated and the distance traveled. The former depends largely on tractive power. The character of machine drawn affects the distance traveled by governing somewhat the number of stops. Stops, however, are due more often to the necessity for taking supplies or making repairs.

22 The steam tractor formerly required a large amount of time for taking on supplies, but it is now easy to take water on the move, and, the transfer of coal, if sacked, is a small item. The internal-

combustion tractor can usually travel a much greater distance without replenishing supplies than the steam tractor, though apparently little effort has been made to balance the capacity of fuel and water tanks. Were it possible entirely to empty either tank, nearly every internal-combustion tractor could travel 10 to 15 mi. under full load without stopping, while the average steam tractor requires a supply of water approximately once every 2 mi. and of coal every 5 or 6 mi.

DAILY CAPACITY OF TRACTORS

23 Sixty steam plowmen in the northwest average 1.48 mi. of furrow travel per hr., and 100 in the southwest average 1.5 mi. With internal-combustion tractors these were increased to 1.73 and 1.68 mi. per hr. respectively. These figures take account of all delays. Actually the traveling speeds are much higher. In one hauling contest, with no stops except for supplies, seven gasolene tractors netted 2.39 mi. per hr., or 92 per cent of the rated high speed, which averaged 2.59 mi. Four steam tractors netted 2.2 mi., or 86 per cent of the rated high speed. The previous year over the same course the gasolene tractors averaged 2.99 mi. per hr. One farm truck maintained 4.42 mi. per hr. with a load 50 per cent greater than its own weight. The same tractors which averaged 2.99 mi. per hr. in a nonstop haulage test averaged only 2.04 mi. of furrow travel per hr. in a plowing contest. The furrows in this case were short necessitating many turns. Each turn takes from 45 sec. to 2 min., hence the net furrow travel was cut to 68 per cent of the hauling speed. Last year, with a longer course (120 rods) six gasolene tractors in plowing averaged in net furrow travel 74 per cent of the rated high speed and 80 per cent of their hauling speed.

24 This year, on a run of practically one mile, 97 per cent of the travel of one tractor was in useful work. The internal-combustion tractors this year averaged 1.86 mi. of furrow travel and the steam tractors 2.22 mi., in tests averaging about five hours. These are not far from the actual traveling speeds in difficult plowing, as there were few stops for supplies or turning.

COST OF OPERATION AND PRODUCTION

25 Comparisons of cost of operation are usually unsatisfactory, owing to the dominating effect of local conditions, and the personal

element. Anywhere from 5 to 20 h.p-hr., may be needed to plow an acre. Ordinary loam shows a resistance of from $4\frac{1}{2}$ to $5\frac{1}{2}$ lb. per sq. in. of cross-section of the furrow slice, say 360 lb. for a furrow 6 in. by 12 in. In plowing an acre of this soil, the 12-in. plow will travel 43,560 ft. and require 7.9 h.p-hr. of work, in addition to the turns, etc. At Winnipeg it required from 12 to 19 h.p-hr. to the acre, plowed 4 in. deep. The transportation of supplies is a highly variable factor. In fact all factors are, and after trying for two years to secure dependable averages, I was forced to generalize in a report dealing with the situation.

26 The manager of a noted Dakota farm this year puts the cost of producing an acre of wheat with horses at $8.45. A traction farmer in the same state produced a 2000-acre crop of flax last year for $6.56 per acre, allowing for all overhead charges. Roughly speaking, the

TABLE 4 COMPARATIVE COST OF PRODUCTION

Cost of Production per Acre of Wheat	With Horses	With Tractor
Land rental	$2.00	$2.00
Plowing	1.35	0.76
Seed	1.13	1.13
Pulverizing and seeding	0.63	0.17
Twine and cutting	0.75	0.39
Shocking	0.22	0.22
Threshing	0.65	0.65
Machinery costs	0.62	0.67
Hauling	1.00	0.26
Incidentals	0.30	0.30
Total	$8.65	$6.55

tractor cuts ten cents from the cost of producing a bushel of wheat in a 20 bushel crop. Table 4 summarizes the comparative cost of production with horses and an oil-burning tractor for conditions in eastern North Dakota. Overhead charges on prime mover are included in the several costs of operations. Machinery costs for the tractor are a trifle higher because of the added investment in suitable plows.

27 The cost of keeping animals is increasing. The scarcity of lumber is making buildings for shelter more costly, Labor is higher in price, as is horse feed. The average farm horse gets 10 lb. of food ($6\frac{1}{2}$ lb. of hay and $3\frac{1}{2}$ lb. of grain) for every hour he works, and some pasturage beside. His thermal efficiency is around 6 per cent when worked ten hours a day and skilfully fed, but ordinarily only around 1 to 2 per cent. Much of his work is light and he probably returns

not over 500 h.p-hr. per year for the $100 which it takes to keep him. Other advantages than cost have given the tractor preference over the animal to date, but as the transition continues the comparative cost will vary increasingly in favor of mechanical power.

In the discussion of the paper Dr. Charles E. Lucke described the contest for tractors held yearly at the Winnipeg fair, and the tractors present there in 1910. The talk was profusely illustrated by lantern slides showing views of the different tractors. These views were published in the Engineering Magazine for September 1911.—EDITOR.

No. 1310

LETTERS PATENT FOR INVENTIONS

A DISCUSSION OF THE RIGHTS GRANTED THEREBY, AND THE PROTECTION THEY AFFORD

By D. Howard Haywood, New York

Member of the Society

There is much confusion in the mind of the average man as to the nature of the right or rights conferred by letters patent of the United States. Upon their face they grant to the holder, for a specified term of years, "the exclusive right to make, use, and vend" the invention claimed therein. These words, taken in their ordinary meaning, would seem to grant to the owner the right to make, use, and vend the invention, coupled with the right to exclude others from so doing. But this, though very commonly accepted to be their meaning, is not their meaning at all. Letters patent grant no right to make, use, or vend an invention, but only the right to prevent others from doing so. The so-called exclusive right is merely the right of exclusion. If the patentee has the right to make, use, and sell the invention, at the time he receives the patent, then the patent grant makes that right an exclusive one; but if he does not have that right at such time, the patent does not give him such right, but merely the right to exclude others therefrom.

2 Take a concrete example. Assume A to be the original inventor of the steam engine; he has a natural right to make, use, and vend the same, entirely regardless of any patent right; he, however, applies for and obtains a broad patent thereon, and thus receives from the government the legal right to the exclusive exercise of this natural right, that is to say, he receives the right to exclude others from exercising the right which they would otherwise have had of making, using, or selling steam engines. Now assume that B at some time later invents a specific form of rotary steam engine. For

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Presented at the New York Meeting (May 1911) of The American Society of Mechanical Engineers. For further material on Patent Law, see The Journal for July 1912.

the novel features therein he is entitled to a patent, and obtains it. At the time of obtaining the patent, however, A's patent is in force and hence B has no right to make, use, or vend a steam engine of any kind. B's patent gives him no right in this connection, for if it did the effect would clearly be in abrogation of A's rights of exclusion, already acquired. B's patent grants him the right to exclude every one from making, using, or vending the specific form of rotary engine invented by him, and nothing more; he can prevent others including A himself, from making, using, or selling the rotary steam engine, but having, at the time he made his invention, no right to the exercise thereof, he obtains none in his patent. It is to be noted also that B's right to exclude A from making the rotary steam engine that he, B, invented, is in no way inconsistent with A's patent right, for, as said, the latter was merely the right to exclude others from the manufacture, use, and sale of steam engines of any and all forms, and not the right to make, use, or sell them himself.

3 The result of the foregoing may seem somewhat anomalous, but it is no less true that during the life of A's patent, neither A nor B can make, use, or sell rotary steam engines, except and unless by permission of one from the other. Failing such permission B can only wait until A's patent right has expired, whereupon he will be free to exercise his natural right, and for the remainder of the term of his own patent will likewise be able to exercise his legal right of restraint as against others.

NOVELTY DISTINGUISHED FROM INFRINGEMENT

4 When application is made to the government for letters patent of the United States, a search is made by the government solely upon the question of novelty. The applicant for a patent right describes and claims what he considers he has invented, and the government, through its officials in the Patent Office, then proceeds to search through prior publications, records, patents, and so forth, in an endeavor to find anything upon which such description and claims can be read. If found, B's application is of course refused, for B in such case is not entitled to any patent right. In the example just given, however, there is no disclosure by A of any rotary steam engine, but merely (say) of a reciprocating steam engine, and it being assumed that no disclosure of a rotary steam engine is found elsewhere, the patent sought for by B is granted to him. The fact that A in his application claimed, and in his patent was given, an

exclusive right (or right of exclusion) in relation to all forms of steam engine, is not taken into consideration by the Patent Office at all. The Patent Office officials are searching for the specific thing that B has disclosed and is claiming; they fail to find it, and a patent is therefore granted to B. B being the original inventor of the rotary steam engine is entitled under the law to the right of excluding anyone else from obtaining the benefits of that invention for the term of years for which such patent rights are granted. The fact that his invention infringes A's patent is of no interest to the patent officials, provided it has novelty, and this is the more readily appreciated when the fact is kept in mind that the grant of a patent to B does not carry with it any right or privilege to make, use or sell the invention covered thereby.

PATENT RIGHTS TRANSFERABLE IN WHOLE OR IN PART

5 The holder of a patent right, i.e., the right to exclude others from making, using, and selling his invention, may remit that right to one or more persons, firms, corporations or other legal entities, at will. Having the right to exclude all persons from the exercise of a certain right which they would otherwise have, he may, if he so desire, undertake not to exercise that right of exclusion as against any selected person or persons. There are in general three ways of accomplishing this result: (*a*) by assignment, (*b*) by territorial grant, (*c*) by license. Assignments may be of the entire patent right held by the orginal grantee, or of a part thereof. If of the entire right the situation is simple and clear; the assignee is merely substituted for the original grantee, and the original grantee steps aside in favor of the assignee, who assumes *in toto* every right the original grantee had at the time he made the assignment. There is provision made for recording such assignments, and a statute provides that when recorded within a specified period, they become and constitute constructive notice to all of the transfer of title of the patent. If, on the other hand, the assignment conveys from the grantee to an assignee, in terms, a part only of the right granted by the patent, the situation is not nearly so clear, and, in fact, is very commonly misunderstood. Assume that A, the original grantee, assigns to X an undivided one-half interest in the patent, such assignment being the usual naked one, i.e., unaccompanied by any partnership agreement. A has now broken up the complete right of exclusion and is sharing it with X. But as the right of exclusion

necessarily carries with it the right of remission thereof, it follows that A and X can now, each of them, and the one independently of the other, remit that exclusion so far as anyone else is concerned, and that neither can interfere with the action of the other in this respect. The "exclusive right" or right of exclusion is thus utterly broken up and lost, unless A and X act in concert, which the assignment does not require them to do; for either A or X can only exclude those from enjoying the benefits of the invention such as the other does not release from such exclusion. X, owning an undivided interest in the patent right, can theoretically release everyone from the patent restraint and A cannot prevent him.

6 Now a step further. We have been assuming the assignment of a one-half interest. But too much stress cannot be laid on the fact that in a naked assignment, i.e., one unaccompanied by any restrictions as to partnership agreement, it matters not one iota what proportion of the patent right is nominally assigned. An assignment of a one-hundredth part conveys exactly the same right in this respect as one-half or ninety-nine hundredths. Any undivided part gives complete rights, subject only to similar rights in the other, and the owner by assignment of any fractional part, no matter how small, may break up his part into as many smaller parts, theoretically to infinity, as he may desire, and may assign those parts to an infinite number of people.

7 I am particularly desirous of warning inventors not to assign fractional parts of their patent rights unless accompanied with a partnership agreement, and preferably with an undertaking that no further rights shall be granted without the signature of all the owners; then, and only then, are the parties protected. In any other case an owner of any part of the patent may proceed independently of the other or others, and may grant other rights indefinitely. This situation is little understood and scarcely ever appreciated, other than by lawyers trained in patent matters; yet it is of paramount importance that it be understood. There is, so far as I know, no parallel in the law thereto. There is no analogy in this respect between the construction of patent assignments and those relating to tangible property such as real estate, or goods and chattels, yet most people act upon the assumption that there is, and so valuable rights are lost or jeopardized.

8 But few words need be said in relation to territorial grants and licenses. These instruments do not convey an undivided interest

in the patent right, but on the contrary, something less, and the exact nature of interest conveyed or granted is dependent in each case upon the wording of the particular instrument, the nature of such interest usually being set forth in specific terms therein. For this reason the ordinary rules of ownership apply and no such warning as has been given above in relation to assignments is necessary.

9 Part ownership of a patent right may result from one other cause, viz., joint invention. In such case application must be made in the name of the joint inventors and the patent is granted to them jointly. In general, the same remarks I have made with respect to part ownership apply, for each owner may operate independently of the other, and may grant rights under the patent without the consent of the other.

THE PATENT ITSELF, AND THE CLAIMS THEREOF

10 Letters patent comprise, besides the formal grant, a specification in which the nature of the invention, and the manner in which the same is carried out, is disclosed.

11 The term specification is, in law, applied comprehensively to the description of the invention, the drawings in which, where the nature of it permits, the invention is illustrated, and the claim or claims which follow the description and in which the scope of the patent right is specifically set forth. The descriptive portion of the specification is commonly termed the specification, the word being then used in a more limited or restricted sense.

12 As it is the claims which determine the breadth and scope of the patent protection granted, it follows that their wording is of extreme importance. It is upon the skillful drawing of the claims that the whole value of the patent depends. They should, where the nature of the invention permits, be broad and comprehensive in their terms, so that mere variations in structure, such as will thereafter suggest themselves to skilled mechanics or to other inventors, will fall within them, yet they must not be ambiguous, uncertain or vague, for they would then be liable to be declared invalid by the courts. Clearly the claims must not read upon any previously invented structure, for in such case they would be anticipated thereby and would have no validity, as has often been said, a structure which, if later, would infringe, if earlier, would anticipate. The first requisite, then, in the drafting of a claim, is an accurate

knowledge of what has been previously accomplished in the same or similar lines, a knowledge, as it is aptly called, of the state of the art.

13 The next requirement is a true appreciation of the problem that the inventor has solved. It is not sufficient that one drawing a claim shall merely understand the specific structure for which patent protection is to be acquired, for in such case he would not be in a proper position to distinguish between the essential and non-essential features thereof. As an example of what I mean in this connection, it is possible to conceive an appliance placed upon an engine such as would operate in connection with the valve mechanisms for both the inlet and the exhaust of steam. For convenience the inventor might apply it in this way, yet actually his problem is conceivably completely solved by the application of the device to the means for admitting steam. A failure to appreciate the fact that the application to the means for controlling the exhaust of the steam was mere surplusage, or at best a convenience, might lead to the drawing of claims in a way that would prevent others only from applying the mechanism to both inlet and exhaust valve mechanism, leaving them free to place it upon the inlet valve mechanism alone, and thereby actually to attain the benefits of the invention. It is not only necessary then that one shall thoroughly understand the mechanism when drafting the claims, but as I have said, there must be a full appreciation of the problem that the inventor has solved.

14 A claim drawn with the foregoing requisites in mind will exclude all that has gone before and will be directed to the pertinent features of invention, and the next thing to be considered is that the language of the claim shall be comprehensive enough to cover changes in structure that thereafter may be made, and it is right here that the most difficult part of the work is found. In the previous requisites noted the precise facts may be readily obtained and the drawing of the claims up to this point is only a question of technical skill in the use of language, but the third consideration deals not with facts at the time but with possibilities in the future. A careful and skilful attorney first draws the claims in as good and accurate form as he is primarily able and then, putting himself in the position of an infringer endeavors to conceive of a structure such as would not fall within the terms thereof. If he succeeds, he changes the language of the claims from time to time to correct their defects, and until they finally satisfy him. For any claim upon a mechanical

device it is thus necessary that the attorney be well versed in the mechanical arts, for how else can he conceive of these various possible structures which might be used in substitution of the structure in front of him? As has been stated above, it will not avail him to use vague, uncertain, and ambiguous terms in an effort to include all possible changes, because even though he may succeed in getting such claims allowed by the Patent Office officials, such claims when in the courts will either be limited substantially to what is shown and described, or will be declared invalid by reason of their indefiniteness, vagueness, and uncertainty. The more definite and precise a claim is, the more it is likely to be sustained, and the more valuable it is upon that account provided it covers sufficiently comprehensively the essential features of the invention.

15 It is true that under the doctrine of equivalents, as it is called, the courts will sometimes construe a claim more broadly than its exact terms; for instance, where a spring has been claimed and a weight may be substituted therefor to produce the same result, the mere substitution of the weight for the spring would not be considered to have evaded the spirit of the claim even though it evaded the precise words thereof. But it is dangerous to rely upon the doctrine of equivalents because one can never be sure as to how far the courts will go in applying it. Furthermore, that one thing shall be the equivalent of another it is necessary that the one thing shall do substantially the same thing as the other, and in substantially the same way. If the change will not conform to this test the doctrine of equivalents will ordinarily not apply, and hence it is highly desirable that the attorney draw the claim in such terms as to cover the possible variations of the structure.

16 After broad claims have been drawn such as comprehensively cover the invention generally, in such terms as to include all reasonable variations of the structure, it is then wise to insert specific claims first to the general specific structure shown, and second, to any specific part of the structure such as may be deemed to be of particular importance to warrant. The first set of specific claims are advantageous for two reasons: first, should the broad claims prove worthless at any later time by the later finding of an anticipatory structure, or should they for any other reason be found to be invalid, then these specific claims may prove of great value. Of course, if the broader claims are sustained in the courts, the specific claims are of little value in this connection, but it not infrequently happens that broad claims are later found to be invalid

for some reason or another and then, if the specific claims cover the infringing structure as well as the structure upon which they were originally drawn, a suit brought on the patent may be sustained and the infringing structure enjoined.

17 By the drafting of claims to specific parts of construction, I have in mind that it is quite common in certain arts to find some particular feature that is quite essential to the proper effective operation of the structure as a whole. This particular feature is often applicable to many and varied structures, and while it may be used in combination with the general structure shown in the application which we are considering, it may be capable of uses in combination with structures such as cannot possibly be included in the other claims to which we have reference. A specific claim to this specific thing will then be of great importance. I will not take up herein the question of whether or not such a claim should be included in this application or indeed could be included under the Patent Office rules in the application to which I have been referring, or whether it had better, or would have to, be included in a separate application. I am rather considering the broad question of what claims would give protection, and from this viewpoint specific claims should be drawn to such specific feature, and indeed to any specific features which may be considered in this connection to be of sufficient importance to warrant it.

WHAT A PATENT MAY COVER

18 Letters patent for inventions may cover: first, an apparatus, machine, structure or a device; second, a process or method; third a composition of matter, such as a chemical compound; and fourth, a design.

19 Under the first heading come such combinations of elements as are included, for instance, in steam heating systems, steam engines, automatic machinery, structures, jewel settings, etc. Included therein is practically everything which comes under the term mechanical arts. Claims drawn to these structures are in form for a combination of elements. The writer is aware that it is common to speak of some claims as basic or fundamental, and other claims as combination claims, with the understanding that combination claims are narrow claims, while basic or fundamental claims are broad ones; but in the last analysis all claims are to a combination of elements, for it is practically impossible to conceive of a claim having but one element therein.

20 The Supreme Court of the United States has laid down the doctrine that the elements of claims must combine together to a single unitary result in order that the claims be patentable. If the elements do not combine together to a single unitary result they are said to be merely aggregated, and an aggregation is unpatentable and the claims drawn thereto invalid. The usual way of expressing this is that a combination is patentable while an aggregation is not. The most famous case perhaps in this connection is the Faber pencil case in which a claim covering the ordinary form of lead pencil in use today, with the rubber tip thereon for erasing purposes, was held unpatentable because for an aggregation. A patent was actually granted upon this device, but the Supreme Court of the United States declared it to be invalid. The Supreme Court said in effect that the pencil and the rubber each operated as independently of the other as each had theretofore operated, and that the mere assembling of them together was for convenience only, and did not constitute an inventive act. When so aggregated the pencil was still used as a pencil and the rubber as an eraser; the pencil and eraser did not combine together for any single unitary result, but on the contrary were separately employed for independent and individual results. For a claim, then, to be valid all of its elements must combine together for a final result which is different in kind from that which would have resulted from the use of these elements individually or in individual sub-combinations. If the complete claim will stand this test it is for a combination and is valid in that respect; if not, it is for an aggregation and will not be upheld.

21 The writer has seen it stated that if the final result attained is the *product* of the functions of the elements in a claim, the claim is for a combination, while if the final result is the *sum* of their functions the claim is for an aggregation; but while this is a terse way of expressing it, it seems almost necessary to understand primarily the difference between a combination and an aggregation when applied to claims before the statement can be appreciated.

22 Under the second heading come processes or methods, and in this connection it is quite difficult to say with any degree of certainty what is capable of being protected by a patent right, and what is not. Anything new in which there is an elemental or chemical action is capable of this protection; while going to the other extreme, anything which is the mere function of a machine is not capable of such protection, the only possible way of protecting the process or method carried out in a machine being by patenting the machine

itself. There is a middle ground, however, in which the question of patentability is always debatable. Upon this middle ground are what are known as step-by-step processes, and of recent years the courts have been leaning toward sustaining this class of method patents.

23 The requirement for patents under the third class, other than novelty, is that they constitute something more than a mere mixture of known ingredients, with no new result other than that which would naturally follow from such mixture.

24 Design patents are granted for something new in ornamental configuration. Parts that have a new form merely for some mechanical purpose are not capable of protection under the design branch of the patent law, the requirement being novelty in ornamental configuration without regard to utility. Novelty in ornamentation and configuration of an inkstand, for instance, would constitute patentable subject matter as a design, while a novel shape of locomotive frame designed for the mere purpose of receiving the parts to be attached thereto better or more conveniently, would not be thus patentable.

FOREIGN PATENTS

25 Letters patent in foreign countries are similar to letters patent of the United States in that they grant no right to make, use, or sell the invention covered thereby, but only the right to prevent others from so making, using, and selling it, but in almost all other respects they differ widely therefrom. They not only differ widely from United States letters patent, but also one from the other. This is true both as to their form, the procedure necessary for obtaining them, and what is necessary to be done in order that they be maintained in force.

26 I do not think it proper or expedient to enter in this paper into a wide discussion or explanation of foreign patents, but certain points will, I think, be of interest and may be touched upon.

27 The first one to be noted is that in almost all of the principal foreign countries, Canada being a notable exception, application for letters patent must be made before there is any public knowledge of the invention whatsoever. In the United States there may be public knowledge for any time not exceeding two years prior to the date of the application, but public knowledge for one day preceding the filing of an application in most countries abroad will prevent the granting of the patent if knowledge thereof is brought to the

Patent Office officials, or will invalidate the patent if granted. Canada allows one year after the issuance of the first patent in any other country, within which application may be made in Canada, but, as I have said, Canada is a notable exception. It will thus be seen that when foreign applications for letters patent are to be made it is vital that the inventor and his associates be particularly careful not to permit a description of the invention to be published in technical journals, not to permit any advertisement to appear such as will disclose the invention, and not to permit the invention to be used in such a public way that it may become generally known, prior to the filing of such foreign applications.

28 I may here make reference to an international treaty known as The International Convention for the Protection of Industrial Property, which was concluded at Paris in 1883. Under the terms of this treaty, applications for patent may be made in any country which is a party thereto within one year from the date of the earliest application (not grant) in any other country, regardless of whether there has been any publication or public knowledge in the meantime. When the latter application is made, the date thereof is carried back to the date of the earliest application elsewhere, the application being treated exactly as if it had been made upon such earlier date. There is a disadvantage to the applicant in this, in that in most foreign countries the letters patent commence to run from the date of the application, so that the effect of the filing in this way, under the terms of the international convention, is to shorten the life of a patent. This, of course, is far outweighed by the advantage of being able to overcome a publication, or a patent applied for previously by another applicant, but subsequent to the date of the earliest application elsewhere by the applicant now seeking protection.

29 The second difference to be noted is that in many foreign countries, Canada being again a notable exception, the applicant for the patent right need not be the inventor. It is sufficient that he be in possession of the invention and is the first to apply for a patent thereon. In the United States the patent right may be granted to an assignee, but the application must be made by the actual inventor, and this also is true in Canada, but in many other countries the name of the real inventor need not appear at all.

30 The third point of difference to be noted is that when once a United States patent has been issued there is no further payment due to the government thereon. In most foreign countries the

patent is taxed from time to time during its life. This tax is relatively small in some countries, while in others it is considerable. Again in some countries it is a fixed amount at stated intervals, while in other countries it is a progressively increasing amount at stated intervals, such, for instance, as yearly. If these taxes are not paid as they become due the patent privilege ceases, so that while in the United States a patent is granted unconditionally for a term of 17 years, in most foreign countries the patent is granted for a varying term of years conditioned upon the prescribed taxes being paid as they fall due, and upon the compliance with other requirements such as the industrial working thereof, to which I will now refer.

31 The fourth essential difference to be noted between United States and foreign patents is that in most foreign countries there is some provision whereby it is required that the invention be industrially worked in the country granting the patent, within a specified term of years, and as a rule continuously thereafter, while in the United States there is no such provision or requirement. In some countries, of which Canada is an example, the patent privilege automatically ceases if the invention be not commercially worked in the country before the expiration of a specified period, while in other countries provision is made that upon application a patent may be judicially annulled if there is no adequate commercial working after a stated period has elapsed. Closely in line with this requirement is a provision in many countries that importation into the country after a specified period will be cause for the termination of a patent privilege.

SUMMARY

32 My purpose in bringing the foregoing to the attention of the members of The American Society of Mechanical Engineers is of a twofold nature. First, I thought that the matter contained therein would be of general interest to a body of men, a large proportion of the members of which are themselves inventors, or are interested in inventions; second, I wish to give inventors and those interested in inventions the following warnings:

a Not to take it for granted that the possession of a patent entitles the patentee to make, use, or sell the invention protected thereby.

b Not to assign undivided portions of a patent right unless such assignment be accompanied with a partnership agreement and preferably with an agreement providing that no further rights be granted, or assignments given, except over the signatures of all of the owners of the patent.

c To see that the claims in their patents adequately and properly cover the invention.

d Not to permit any publication or public knowledge of the invention to take place prior to the application for foreign patent where foreign patents are to be obtained.

33 It is my experience that the failure to procure proper and adequate patent protection, or to retain it should it have been acquired, is in a large proportion of the cases directly traceable to a want of understanding or comprehension of the foregoing, or to a direct disregard of such warnings as I have given.

SYMPOSIUM ON CEMENT MANUFACTURE

No. 1311*a*

SOME PROBLEMS OF THE CEMENT INDUSTRY

WALTER S. LANDIS,[1] SOUTH BETHLEHEM, PA.

Non-Member

It would probably be assumed by the average mechanical engineer that the cement industry lay in the field of chemistry. While it is true that in the early days of the industry the chemist occupied the most important position, the mechanical engineer is at the head of our great plants and the chemist has degenerated into a very subordinate official. There are reasons for this change, the most important one being that it is easier for the mechanical engineer to pick up the necessary chemical knowledge than it is for the chemist to acquire the requisite mechanical training. Again, the rigid requirements as laid down by the early chemists for the manufacture of a passable product are no longer recognized as true since we have learned so much more about the technology of cement. As a result we have been having the development of the plant as against the process, and the mechanical engineer, of course, plays the most important part in such plant development.

2 What we today understand as Portland cement is a certain compound of silica, alumina and lime in the proportions of SiO_2, 20 to 23 per cent; Al_2O_3, 8 to 10 per cent; and CaO, 62 to 65 per cent. It is not possible to obtain these ingredients in large quantities in a state of absolute purity and we frequently find alumina replaced by ferric oxide, up to 3 per cent, and lime by magnesia, up to 2 or 3 per cent, so that actual cements as found on the market diverge slightly from the above analyses without, however, suffering greatly in their phys-

[1]Associate Professor of Metallurgy, Lehigh University.

Presented at the Spring Meeting, Pittsburgh 1911, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

ical properties. It must also be remembered that the well-known Portland cement is only one of a dozen chemical compounds which harden or set when mixed with water and that it has attained its great importance because of the ease and cheapness with which it can be manufactured and not because of any peculiar properties not possessed by certain other compounds.

3 Simply the mechanical mixing of silica, alumina and lime in the above proportions will not yield a compound possessing the properties of Portland cement. It is essential that these ingredients be in a manner combined, not exactly as a true chemical compound, but rather as a physico-chemical solution of one or more chemical compounds in each other. The mixture must be finely ground before it will exhibit the characteristic setting property. The best and practically the only way in which such a union can be attained is by a complete or partial fusion of the silica, alumina and lime mixture. Such a partial fusing is called sintering or clinkering.

4 All the laws of physical chemistry relating to the formation of slags and fused mixtures apply also to the formation of the clinker, a few of the more important facts being here mentioned. The melting point and the clinkering temperature are dependent on the purity of the mass clinkered, being lowered by the presence of such impurities as oxide of iron, magnesia, alkalies, etc. It is a well-recognized fact that the so-called white or stainless cements require a much higher temperature for clinkering than the dark colored and more impure mixtures. The temperature required for clinker formation depends on the intimacy of the contact of the various clinkering ingredients; the finer the individual materials are ground the lower the temperature required.

5 Since we know that such a thing does not exist as an overburned cement, a fusion of the properly proportioned ingredients would solve the question of a homogeneous product. But such a fusion has not proved advisable in practice and the cement manufacturer has had to content himself with a sintered or clinkered product, that is, one which has been raised in temperature to a sticky or viscous stage and not actually liquified. As such clinkered product is at no time a mobile liquid we cannot depend on liquid diffusion for the proper mixing of the silica, alumina and lime but must do such mixing purposely. To insure the necessary contact between these ingredients so that they can unite chemically to a homogeneous whole when in the state of fusion which is only incipient, they must be most finely ground and most thoroughly mixed. Therefore the first stage in all

cement processes, as at present practised, is a preliminary grinding and mixing of the raw materials.

6 The first consideration in the grinding of a material is a study of the properties of the material to be ground. This in turn leads us to a description of the raw materials entering into the manufacture of Portland cement. Of first importance among these is cement rock, an argillaceous limestone, lying both in composition and in geological position between the true limestones and the slates. It is a natural mixture of carbonate of lime and clay, which on heating to the clinkering temperature loses its combined water and carbonic acid and unites together into a fritted mass of the desired composition and properties. It is shaly in nature, not very hard, and of varying degrees of toughness. In several favored parts of the country, as for instance one or two places in the Lehigh Valley region, this rock needs no further treatment than quarrying, comparatively coarse crushing and burning, to form a first-class clinker. Usually the ingredients are not present in the natural rock in exactly the desired proportions and the one lacking, generally lime, must be added. Sometimes in this region there will be two strata or benches in the same quarry so constituted that by mixing rock from the two in certain proportions, the silica, alumina and lime will be in the desired proportions in the clinkered product.

7 Where materials are found to be so near the desired composition as to require practically no mixing, the grinding is a simple proposition. Only such fineness would be required, if the rock were of uniform composition, as would insure the kiln producing a properly burned product when run at a predetermined capacity. This would really be a sizing rather than a grinding. On the other hand, if the rock is non-homogeneous or requires some mixing to make a properly constituted clinker, then finer and finer and finer grinding, proportionate to the degree of non-homogeneity or amount of mixing required, must be done to enable the mixture to attain the composition desired in the clinker. No hard and fast rule can be laid down for the required fineness of grinding. It must be suited to the individual case, remembering that sometimes the financial success of the plant depends on making a passable product with the coarsest allowable grinding. The kiln temperature, as well as the length of time the mass is subjected to this temperature, influences the required fineness of grinding. The finer the grinding the easier the clinkering and, therefore, the capacity and coal consumption of the kilns are directly proportional to the fineness of the ground mix fed in. It has been found in practice within the

past few years to be more economical to grind very fine before burning, even though such a degree of fineness of the raw material would not actually be required to make a satisfactory clinker. In other words, the coal used to drive the grinding machines to produce this extra fineness is less than that saved in the kiln, and at the same time the capacity of the kiln is greatly increased. It has also been said, though I cannot verify it from actual observation, that the finer the grinding of the raw material the easier the clinker is to grind afterwards. This might be explained by the fact that the lower temperature at which the finer mix clinkers leads to the formation of a softer clinker. In that case the power put into the preliminary grinding would at least be partially saved in the finishing department. A more thorough study of this question is one of our problems.

8 But it must not be understood that the manufacture of cement is confined to those favored localities in which cement rock is found. A mixture of clay and limestone can be prepared artificially of such composition that when the combined water in the clay and the carbonic acid of the limestone are driven off by heating, the residue will form clinker of the desired composition. Here we have materials of entirely different composition and nature to be mixed, and the grinding of each must be exceptionally fine in order that they may combine under ecomonical conditions of kiln running. It is recognized by cement men that the manufacture of a satisfactory cement from limestone and clay entails a different mill equipment from that in use in the Lehigh Valley region, where the raw material is a nearly perfect cement rock. Such raw materials, to be properly mixed, *must* be ground so that practically all will pass a 100-mesh screen and nearly all a 200-mesh.

9 Other raw materials available are marl mixed with limestone or clay, shale mixed with limestone, and even blast-furnace slag mixed with limestone. This latter material is at present receiving a great deal of attention from cement men. The iron blast-furnace slag is composed of silica, lime and alumina, and by the addition of limestone, clinker of the desired composition can be produced. It has the advantage of being a waste product which the furnaces are glad to get rid of and at the same time needs no quarrying or coarse grinding, since it is granulated in water when tapped from the blast furnace. All materials like clay and limestone require exceedingly fine grinding (all through 100 mesh, 50–70 per cent through 200 mesh) in order to insure proper combination in the kiln.

10 It is not possible to discuss in detail the many problems connected with fine grinding of the large quantities of rock and clay which

the cement industry handles. In quarrying the rock the percussion drill, such as is used in the digging of ordinary drilled wells, is largely used for making the blasting holes, a series of holes being drilled along the face of the quarry 15 to 20 ft. back. The author has seen single blasts bring down from 20,000 to 40,000 tons of rock. After blasting, the rock is loaded up into dump cars in many quarries by the use of a steam shovel. These cars are moved to the foot of the hoisting incline by gravity or mule power, and are pulled up by a donkey engine. Much engineering skill must be brought to bear on the economical handling of the quarried rock in the quarry itself. After arriving at the top of the quarry the car is run into the crushing house, where the stone is weighed and dumped into the first crushing unit.

11 American and European practice differ in the size of this first crusher, which is almost universally of the gyratory type. With us, hand labor is expensive and must be dispensed with wherever possible. We therefore build this unit as large as is required to take any single piece of rock which the steam shovel can load into the dump car. When such a carload is dumped into the crusher the driving motor of the crusher suffers a momentary peak-load as the rock goes through, which load then drops off to nothing until the next car arrives. These crushers sometimes take a piece of rock 30 to 36 in. in diameter, using at the peak 250 h.p. In Europe, the rock as quarried is broken much smaller, usually by hand sledging, thereby permitting the use of a number of smaller gyratories, and so distributing the load on the motor more uniformly throughout the day. In our country the first large coarse crushers are followed by smaller ones of the same type, and in the newer mills these are driven from the same power source as the larger ones.

12 The advisability of using the very largest types of crushers made is a much discussed point among cement men. When one considers that the very large pieces of rock quarried are the exception rather than the rule and that the crusher is rarely called upon to take one the size of its opening, it seems questionable to fit a mill with such a machine. A stick of dynamite laid on such a large piece will easily reduce it to a size readily handled by the smaller crushers (even in this case larger than the European crushers), and so eliminate the enormous peak loads and expensive machinery. Several large new mills have not provided themselves with the largest procurable type of crushers made, even though their output would warrant it. This question of crusher size is another of the problems of cement manufacture awaiting final solution.

13 The drying of the product of the gyratories is the next stage in cement manufacture. The fineness to which the material is to be

ultimately ground makes it necessary to remove all hygroscopic moisture, otherwise clogging of screens and cutting down of capacity of the fine grinding mills would result. The driers are cylindrical steel shells capable of being rotated on their axes and are set slightly inclined in order to insure progressive motion of the charge through them. The wet, coarsely crushed material is fed in at the upper end and hot gaseous products from a coal fire or natural gas flame pass in at the lower end and over the charge to be dried. They are frequently provided with buckets, formed of Z-bars riveted to their inner surface, to pick up the rock and carry it part way around and so drop it through the gases passing through the drier and at the same time advance it through the kiln. At times they are divided into compartments to insure closer contact between the heating gases and the rock, but this design is gradually dropping out of use. The great decrease in capacity of a compartment drier over the ordinary type more than offsets the greater heat transfer from the gases to the feed. It is only recently that attempts have been made to utilize the hot gases from the kilns to perform the drying of the mix. Several such installations have recently been made and are working with great success.

14 The drying of clay for fine grinding presents an unusual problem, because after being dried it must be used immediately, as it again takes up moisture from the atmosphere. This usually results in a double drying operation: first just before disintegration, after which it goes into storage, and a second drying after lying in storage and just before final grinding.

15 From the driers the rock passes to one of the various types of fine grinding mills, such as the ball and tube mills, or mechanical mills like the Griffin and Fuller. The excellent catalogues furnished by the manufacturers of this class of machinery make a detailed description unnecessary here and reference is made to them with full confidence in the reliability of the data there found. The product leaves these mills so finely ground that more than half of it will usually pass a screen of 200 meshes to the linear inch. It has been universal practice to fit the mechanical mills with screens (no screens in the tube mills) of 30 to 40 mesh openings, through which the product of the mill has to pass before discharge. Such screens are placed so that the material discharged from the mill strikes them at an acute angle and the full size of the opening is never available to the discharged product. Recently air separators have been tried as a substitute for the expensive and troublesome screens. The success of this installation is doubtful, for the conditions under which air separators work most

efficiently are not the conditions under which ground material should be prepared for the cement industry. A word of explanation will make this clear.

16 The screens having been entirely removed from the mill or replaced by much coarser ones permit a continuous supply of partially ground material to pass to the air separator for sizing, say to 100 or 200 mesh as desired, the oversize being returned to the mills for further grinding. With proper operation of such an air separator very little of the material will be ground finer than the air separator is adjusted to sort out. Now it is a well-known fact that the finer the materials entering into cement manufacture are ground, the better the product made. Since air separation furnishes a very uniformly sized product, this desirable extra fine grinding is eliminated. The screens, on the other hand, offer difficulty to the passage of product out of the mill, keeping a large part of the charge under grinding influence for some time after it has reached screen size, reducing it finer and finer in size. Thus some of the product discharged from a mill having 30 to 40 mesh screens may be 500 or 1000 mesh in size, a condition not realized in highly developed air separation. It is, however, admitted that air separation leads to greater capacity of output of the grinding unit, though at the expense of quality.

17 So far nothing has been said of the mixing of the raw materials to insure proper composition of the finished cement. This mixing practice has varied greatly. Originally the several ingredients were ground separately, mixed with water until thin enough to flow readily, and streams of each run upon a drying floor. Their fluidity insured mixing. After drying for some time, the mass was rolled into balls or pressed into bricks, and stored until ready for charging into the clinkering furnaces. Today this practice is obsolete and in its place we have the two great systems under which modern mills operate. They are known as the wet and the dry processes, their chief differences being based on the operation of mixing. In the wet process the two ingredients are ground separately, mixed with water until thin enough to flow, pumping assisting in this conveying if necessary, and are charged in approximately the desired proportions into large agitating tanks for mixing. The final adjustment of the composition is made in these tanks. The wet mixture is then pumped from these tanks directly to the kilns.

18 In the dry system the properly weighed and proportioned materials are first separately crushed in the coarse crushers, and are fed together into the fine grinding mills, practice varying somewhat as to

the exact mill into which the two materials first come into contact. The operation of grinding performs all the mixing required. For the same material it is supposed that finer preliminary grinding is more necessary in the dry process than in the wet for the same quality of cement. It seems to the author useless to add 50 to 100 per cent of water to a mass of ground material and then to evaporate that water again, when the dry process is working so satisfactorily.

19 A good many changes have been made in the arrangement of the preliminary or raw grinding side of the mills as the industry developed. It is expensive to handle material and to transmit power. In the older days of the central power plant and lineshaft drive, the mills were arranged solely with a view to economy of power transmission and much handling of the material between the various crushing and grinding units was the result. Today the individual motor drive is supreme, with the result that I have seen modern mills so laid out that there is scarcely any handling of material in the whole grinding department, the first elevation occurring into bins above the kilns. Gravity was used to feed from one unit to the next. Such an arrangement is not always the most advantageous when it comes to a breakdown; and the mill must be built on a hillside, which means awkwardness in handling repair parts. The ideal mill would probably require at least three elevators in this department, and since the power required by such elevators would be hardly over two or three per cent of the total power installation of the plant, it is really not a serious item.

20 Another feature of mill design receiving attention is storage capacity. The rough handling of the machines causes them, rugged as they are, to be laid off for repairs. It is necessary, to insure running of the mill to provide ample storage capacity between the various classes of grinding machinery so that a stoppage of one part will not cause the shutdown of the entire mill. Several very recent mills have provided storage capacity as follows: a 4-day supply of gyratory discharge, a 12-hour supply of ball mill feed, a 12-hour supply of finishing mill feed and a 24-hour supply of kiln feed. It must be remembered that climatic conditions have a marked influence on the consideration of storage capacity.

21 The kilns originally used for burning the mix to clinker were fixed vertical, bottle-shaped furnaces. The balls or bricks made by the old wet process previously described were fed in at the top and the burned product drawn out at the bottom. The product was so irregularly burned that hand sorting had to be employed to pick out the under

burned product, this portion containing uncombined lime and not being suitable for the manufacture of a sound cement. While the fuel consumption of such a kiln is much below that of the usual rotary kiln, the labor and attendance charges are very high and the apparatus has been obsolete in this country for many years. Even Germany, a country of cheap labor, is now replacing this type of kiln by our own rotary design.

22 The modern rotary kiln consists of a cylindrical steel shell lined with refractory material and supported on rollers in cradles, its axis being slightly inclined from the horizontal. A rotation on its bearings is secured through one of the many types of variable-speed transmission. Its upper end enters into what is called the chimney hood, a dust trap and chimney support combined, and its lower end into another movable hood adapted for the discharge of clinker and the entry of fuel nozzles for pulverized coal, gas or oil. Pulverized coal is the most widely used fuel in cement burning. In size these kilns vary from 60 ft. long and 5 ft. in diameter up to 240 ft. long and 12 ft. in diameter. The output of a kiln is not easily determined, as it varies much with the nature of the product to be clinkered, the degree of fineness of the feed, the moisture to be evaporated, and whether the driving demands are for efficiency of operation or for capacity. Great capacity is not synonymous with high efficiency in a cement kiln. For example, a 60-ft. kiln in the Lehigh region will have a capacity of 200 bbl. of clinker per day and the 240-ft. kiln of approximately 2500 bbl. per day. The lining of the kiln, at least at the hot end, is either of bauxite or magnesite brick in the majority of installations, though a highly refractory fire-clay may be used. Either of the first two linings is of a much higher heat conductivity than the latter.

23 One of the features of kiln design needing the attention of the fuel engineer is the stack. One sees on the same sized kilns stacks of all heights and diameters and not all can be highly efficient and economical. Some steps should be taken to investigate this question and to formulate a standard design embodying economy and efficiency.

24 Some of the finely ground material fed into the kiln is picked up by the gases passing through the kiln and passes into the chimney. Provision must be made in its design for settling out as much as possible of this dust, and too often one sees a narrow chimney on a kiln carrying out clouds of finely ground material and scattering it for miles over the surrounding country. This material consists of the extreme fines so greatly to be desired in the product.

25 The physical and chemical reactions taking place in the raw

material as it passes through the kiln, starting with its charging at the higher or chimney end of the kiln, are as follows: Rise in temperature accompanied by evaporation of water, if present, such evaporation holding the temperature constant at about 100 deg. cent. in the wet processes; rise in temperature up to about 300 deg. cent. where any chemically combined water such as occurs in clay is driven off, the rise being less rapid during such dissociation; rise in temperature up to about 800 deg. cent., where the carbonic acid of the limestone is driven off, the rise in temperature being again sharply checked by the heat absorption occasioned by this decomposition; a rise in temperature up to that of clinkering, 1200 to 1400 deg. cent., depending on the various conditions existing in the chemical composition and physical properties of the mix, as already discussed; rapid rise in temperature to a maximum, at times higher than the temperature of the flame in the kiln, due to the heat of combination of the ingredients in forming the clinker; rapid drop in temperature as the material is discharged from the kiln. It is not possible to define in feet and inches the length of each zone because of the variable character of the raw material in each cement producing district and of the great differences of kiln length. A specific case was discussed in the paper[1] read before the Society by Mr. Soper.

26 If we consider for the time being the charge of a kiln in the Lehigh Valley region where cement rock and the dry process are used, the net heat required by the above chemical reactions would be furnished by the combustion of about 16 lb. of average long-flame bituminous coal per barrel of clinker formed. This amount would be increased where clay and the wet process are used because of the heat absorbed in evaporating water and decomposing the hydrated compounds. The average fuel consumption of coal-fired cement kilns throughout the country is about 90 lb. of coal per bbl. of clinker, the calorific value of the difference between this figure and the one just given being lost as sensible heat in the discharged clinker and chimney gases and by radiation and conduction from the kiln shell. Waste of fuel of this magnitude should certainly appeal to the fuel engineer as well worthy of attention, particularly if he considers the magnitude of the industry. In 1910 this country made nearly 70,000,000 bbl. of cement. It is only recently that intelligent attempts have been made to reduce the heat losses of the kiln, or to utilize the waste heat in other operations. A feeble attempt has been made in this country to pass the hot chimney gases (temperatures averaging

[1]The Rotary Cement Kiln, by Ellis Soper, published in The Journal for October 1910.

500 deg. cent.) through the driers and so eliminate the extra fuel required in that part of the plant. Such installations are very successful. The regeneration of the heat in the discharged clinker by passing it through an under-cooler and preheating the incoming air has not proved satisfactory in many installations because of the stupidity of the operators. Where properly run, this system is saving hundreds of tons of coal per year. The main trouble has been that the kilns became too hot and the clinker stuck to the sides, all because the kiln manager did not deem it necessary to cut down the fuel when heat units were fed in another manner. German cement manufacturers used both the dryer and the under-cooler with their kilns almost from the introduction of the rotary type. It is only since the publication by the author of figures on radiation from kilns and the means of reducing such losses, that attempts have been made to save fuel by the installation as a backing to the usual kiln lining of a new heat insulating material.

27 It is not impossible to conceive of the use of a different and more efficient type of furnace than the rotary kiln for performing the clinkering operation. Surely fuel engineers have a prize well worth striving for when one considers that a saving of 10 to 15 lb. of coal per bbl. of clinker means a saving of three-quarter of a million dollars annually to the cement industry.

28 The clinker, after leaving the kiln, is passed through a series of cooling devices to dissipate quickly the heat it contains, and is then stored in heaps, exposed to the weather to age. Cement that has been underburned or improperly mixed so that it contains free lime will not pass the boiling test. If allowed to remain in the weather for some time so as to slake this lime and even lose some of it by solution, the cement prepared from such weathered product is generally of improved quality. The aging of clinker renders it softer and more easily ground, increasing the output of the grinding mills as much as 50 per cent. The elevators and overhead devices needed to handle such a storage are not expensive nor complicated. The increased capacity of the grinding machines more than offsets the power required by the conveying and storage machinery, and it is believed that the consequent ability to grind finer also offsets the bad effects of any uncombined lime which may be present.

29 After being recovered from the aging heaps the clinker passes into the final grinding department of the mill. This is similar to the raw grinding department, with the omission of the gyratories and dryers. Gypsum is mixed with the clinker and the mass then passes to the grinding mills to be again reduced in size. As we have seen

that clinker may lie exposed to the weather without taking up water, there must be a maximum size of particle which will combine it with water. This has been variously placed at about 150 mesh size, all particles coarser than that being inert in the finished cement. For this reason it is necessary to grind so that the greater proportion of the product will again pass through the 200 mesh screen. After this grinding the cement passes to the stock house, where it is stored for packing and shipment. It is now packed by automatic weighing and packing machines, in barrels and bags, the latter predominating.

30 The important subject of conveying machinery as applied to the cement mill cannot here be considered. Buckets are used for elevating, air not proving successful; lump material like clinker is carried on belts or by drags; and for fine material the screw conveyor has no great competitor.

31 The power required for driving the mill varies very much with the type of installation. With the old shaft drive it was not infrequent to use $1\frac{3}{4}$ h.p. per bbl. of output per day. Some of the new motor-driven plants use as low as $\frac{2}{3}$ h.p. per bbl. per day, and the average throughout the country is not far from $1\frac{1}{4}$ h.p.

32 With the decreasing profit in cement manufacture the difference between the minimum and the average power requirements as given above should be sufficient to cause a remodelling of power plants and transmission machinery in the older mills. The gas engine should find a growing field in this industry because the individual machines lend themselves so readily to motor driving. The labor requirement of the various mills seems to run parallel with the power requirement. In the older mills 8 bbl. of cement per day per man employed was a good average, while the newer mills are producing 16 bbl.

33 Cost figures are difficult to obtain in any privately controlled and competitive industry, and the cement industry is no exception to this fact. In a recent presidential address before a scientific society the cost of cement production was placed at approximately 10 lb. for 1 cent, or 40 cents per bbl. The author has met with but one cost sheet which in any way compared with this, a sheet representative of the practice of a certain Western mill which soon went into receivers' hands and had been idle a great part of the time. I do not believe that a single mill in the country is putting cement into bags at that figure, for in most it barely covers fuel and labor costs alone. The Lehigh region is producing cement as cheaply as any in the country and we have to strive hard to cut the total cost at the mill below 55 to 60 cents per bbl.; it is nearer 70 cents for the Western mills.

No. 1311*b*

THE EDISON ROLL CRUSHERS

By W. H. Mason, Stewartsville, N. J.
Member of the Society

34 The rapid growth of concrete construction in this country is causing unusual interest in the manufacture of cement, most of which is made from broken stone, a material which also constitutes about two-thirds of the bulk of ordinary concrete. This paper discusses the Edison method of quarrying and crushing stone, both for the manufacture of cement, flux concrete, railroad ballast, macadam, etc., and ore for metallurgical purposes.

35 Some years ago, Thomas A. Edison experimented with the concentration of a very lean magnetic iron ore at Edison, N. J., employing the usual plant for crushing stone, which consisted of several jaw crushers. The stone in the quarry was drilled with a close spacing of drilled holes, and after being blasted was broken up by hand sledging into pieces, approximately 100 lb. in size. This caused a large expenditure for drilling, dynamite, hand-sledging and hand-loading. Mr. Edison soon realized that in concentrating this lean iron ore commercially it was necessary to reduce the cost of quarrying and crushing to a much lower point than had been realized heretofore in operations of a similar character, for in order to produce one ton of concentrates it was necessary to quarry, crush and treat about four tons of the lean ore.

36 In approaching this problem, Mr. Edison reasoned "the total heat or energy in 1 lb. of pea coal is approximately 12,500 heat units, but only about 15 per cent of this, or 1875 B.t.u., is available in mechanical energy through the medium of boilers and steam engines, while the available B.t.u. in 1 lb. of nitro-glycerine is approximately 3650. Therefore, in 50 per cent dynamite there is available 1825 B.t.u. per lb., or the same mechanical power that can be derived from 1 lb. of coal. But 1 ton of pea coal is worth approxi-

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

mately $2.50, while 1 ton of dynamite is worth about $250, making the commercial advantage of the coal over the dynamite approximately 100 to 1."

37 Realizing also that a large part of the dynamite used in ordinary quarrying operations was expended not so much in breaking out the stone from the ledges in the quarry, but in reducing this stone to such sizes as could be handled in the crushers, he set out to design a crusher capable of taking much larger stone. He first constructed a pair of rolls 5 ft. long and 6 ft. in diameter having small protuberances on the surfaces, as shown in Fig. 1. This would take and crush larger pieces than the jaw crushers then in use, but if a stone were fed to the rolls greater than the angle of grip of the rolls, it would ride on top and rapidly wear down the knobs on the plates or tear the plates from the surface of the rolls.

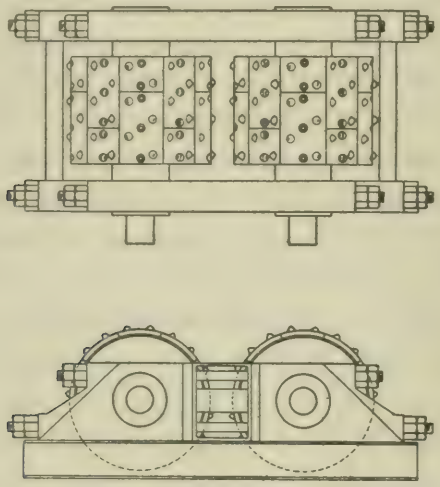

FIG. 1 ORIGINAL ROLLS PATENTED IN 1895

38 Mr. Edison then invented the slugger, consisting of two rows of high knobs on one roll set diametrically opposite (Fig. 2). These knobs were put on with the idea of slugging the larger stones mechanically to such a size as to come within the angle of grip of the rolls. He had found that the rolls would frequently stop on receipt of a large charge of stone, not having sufficient power to crush the rock. To remedy this he ran the rolls at a much higher speed than when they were first erected, storing up sufficient kinetic energy to perform the actual crushing operation. This increase of speed accomplished two purposes: it increased the kinetic energy and delivered a much harder blow from the slugger knobs.

39 When a stone is received by the rolls larger than they can grip, it tends to ride on top of the rolls, but the slugger prevents this, since it delivers 440 blows per minute and the stone must rise two or more inches in approximately $\frac{1}{24}$ of a second or be shattered.

40 The rolls were designed primarily to reduce the expense of quarrying, which is usually much larger than the crushing costs, whatever type of crusher is used. In order to do this, it was necessary to set the drilled holes far apart and to blow out the stone or ore in large pieces, thus making a great saving in both drilling and blasting. Mr. Edison also had two steam shovels especially constructed, much heavier than any previously built, and special cars to load, transport and dump these large pieces of stone into the rolls. With this combination it was possible to reduce the cost of quarrying and crushing to about one-fourth of what it had been when using the jaw crushers. About the time these improvements in quarrying and crushing were made, the development of the Mesaba iron ore deposits caused such a reduction in the price of iron ore that the Edison plant, with its extremely low-grade ore was unable to compete.

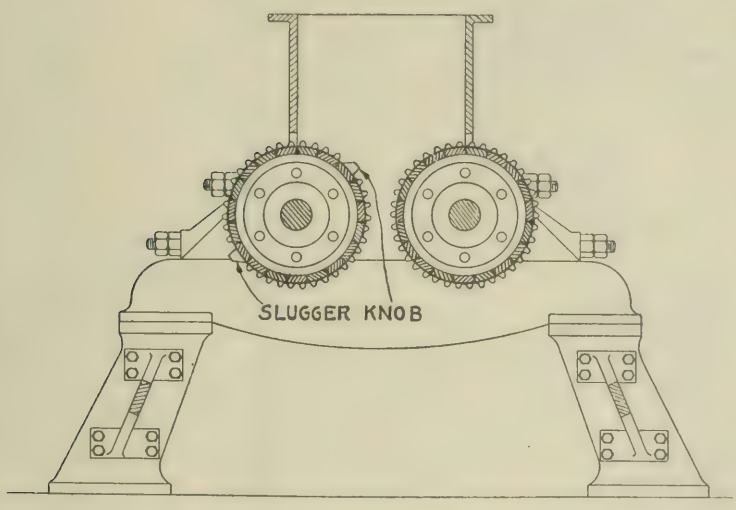

FIG. 2 ORIGINAL ROLLS WITH SLUGGERS ADDED

41 Having developed these crushing rolls and much other machinery for handling and milling ore or stone, Mr. Edison subsequently projected the Edison Portland Cement Company and there installed the set of 5 ft. by 5 ft. rolls shown in Figs. 3 and 4.

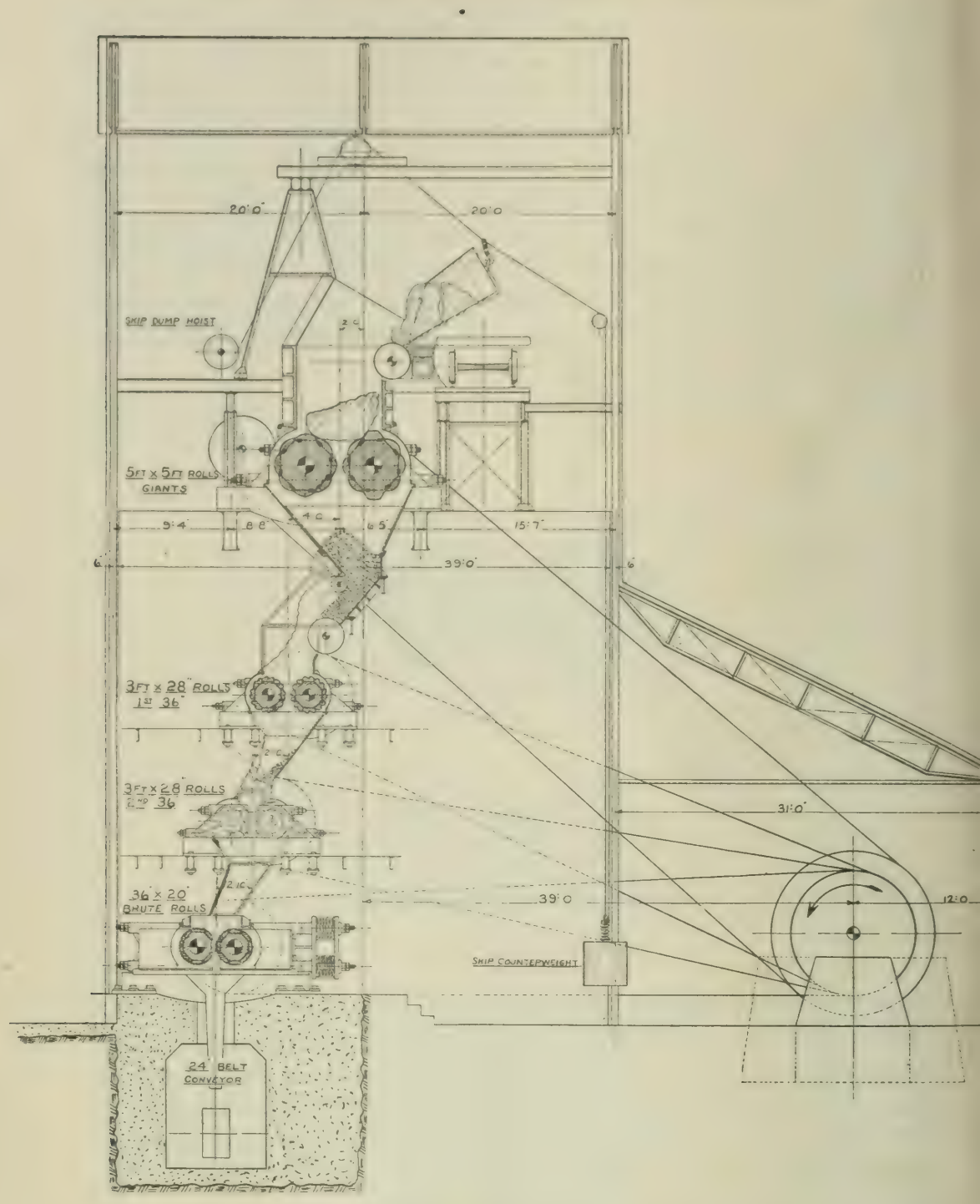

FIG. 3 SECTIONAL VIEW OF THE CRUSHING PLANT OF THE EDISON PORTLAND CEMENT COMPANY

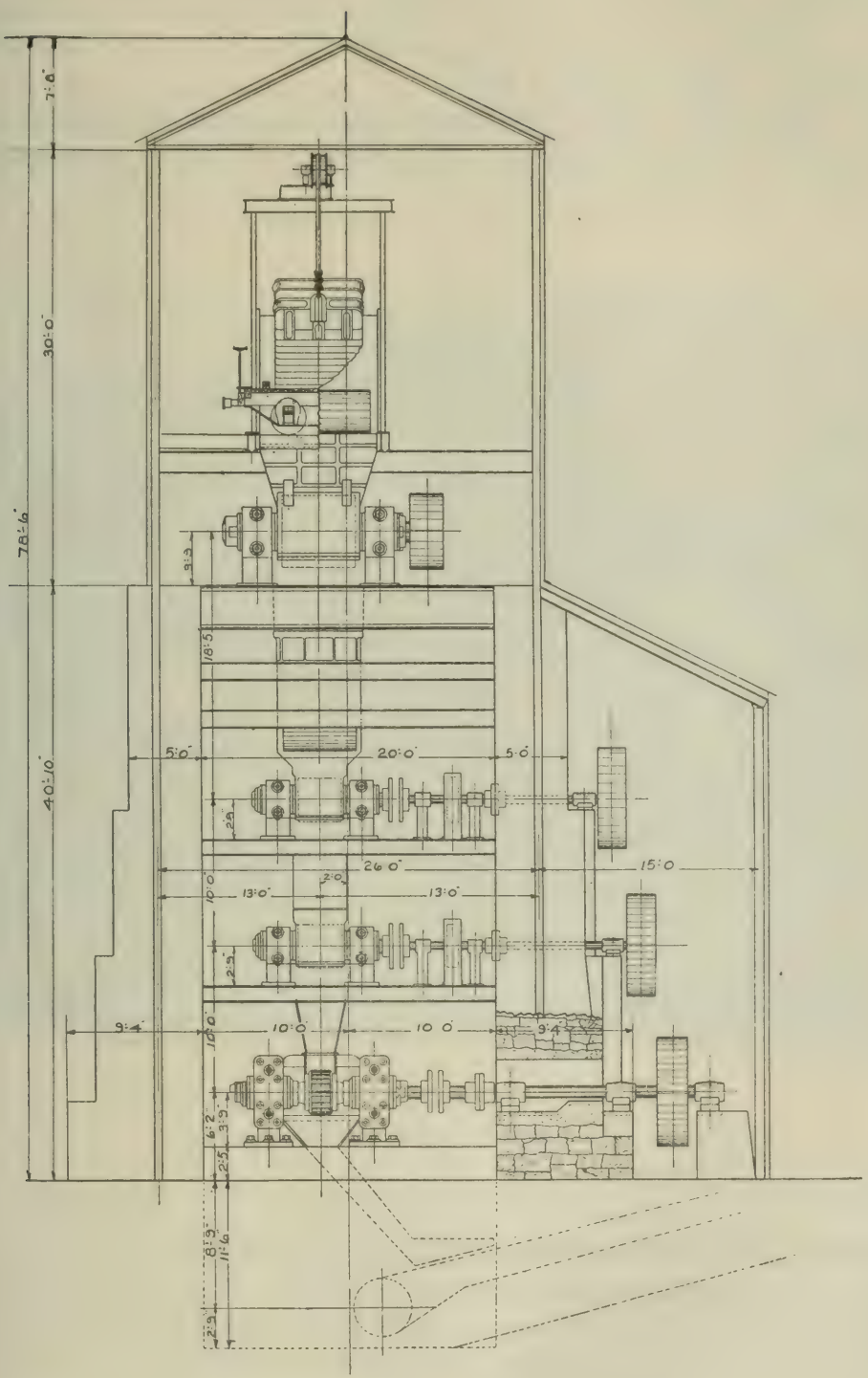

FIG. 4 END ELEVATION OF THE CRUSHING PLANT OF THE EDISON PORTLAND CEMENT COMPANY

These views show the general arrangement of the crusher house, containing 4 sets of rolls, which reduce the stone successively from pieces weighing 8 to 10 tons to $\frac{1}{2}$-in. chips. This crushing plant has been in operation for 8 years.

42 Before Mr. Edison constructed his giant rolls, the largest rolls in use were those known as the Cornish rolls, which were geared

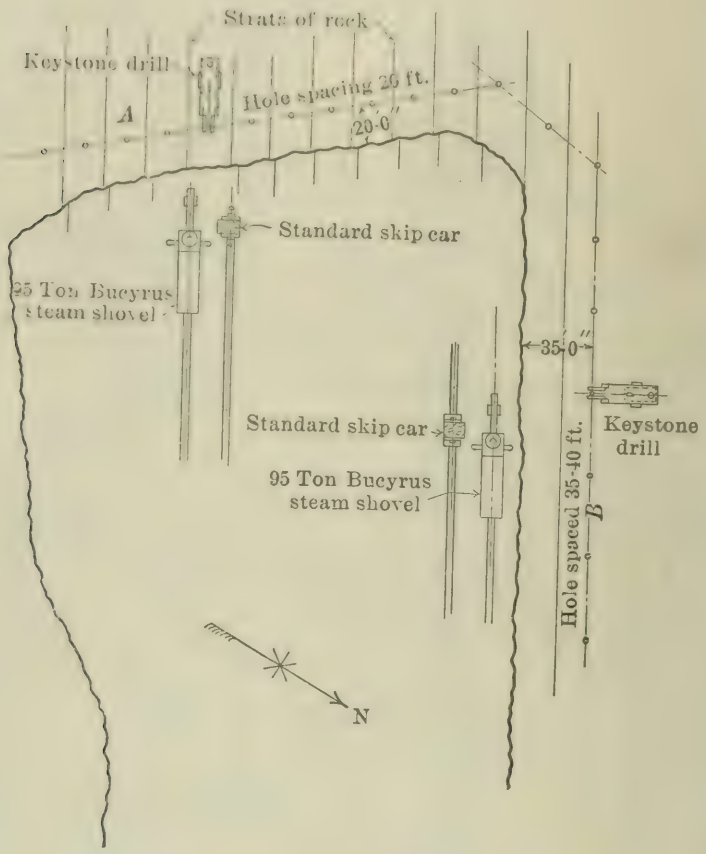

FIG. 5 DIAGRAM OF CEMENT STONE QUARRY

together. The kinetic energy of a pair of these rolls, 16 in. in face, 30 in. in diameter and 100 r.p.m., is 8100 ft-lb. The kinetic energy of a pair of 5 ft. by 5 ft., 220 r.p.m., Edison rolls is 2,280,000 ft-lb. The kinetic energy of a pair of Edison rolls, 6 ft. in. in diameter by 7 ft. long at 175 r.p.m. is 4,217,000 ft-lb. The largest gyratory crusher that the writer knows of has a 42-in. opening; that is, it will take a stone something less than a 42-in. cube, while the Edison 6 ft. by 7 ft. giant rolls will take a stone about 7 ft. cube. The 42-in. cubic stone weighs

approximately $3\frac{1}{2}$ tons, while a 7-ft. cube weighs approximately 28 tons, which is probably larger than can be handled economically by the largest steam shovels now manufactured. It is an ordinary occurrence, however, to crush a stone weighing 15 to 20 tons.

43 The cement stone quarry of the Edison cement plant is an open cut with the strata running vertically or approximately so. The stone rises from 60 ft. to 80 ft. above the floor of the quarry, with a 3-ft. or 4-ft. layer of clay on top, which is removed by a small revolv-

FIG. 6 STEAM SHOVEL LOADING LARGE STONES

ing steam shovel and loaded into carts. Holes are drilled with a churn drill from the top of the quarry to about 6 ft. below the floor. These holes are 6 in. in diameter and about 20 ft. apart and set back about 20 ft. from the face of the quarry, which is worked on two sides, i. e., at *A* and *B* (Fig. 5). On account of the stratification, indicated in the figure, it has been found necessary to use more dynamite when working face *A* than when working face *B*. While working *B* the holes can be spaced at least twice as far apart when using the same amount of dynamite in each hole as in those on face *A*. When a hole is drilled

it is usually squibbed, by putting 30 to 50 lb. of dynamite in the bottom of the hole and exploding it. This enlarges the bottom of the hole, making room for more powder at this point. Then the holes are loaded with 50 per cent dynamite, which fills up the portion which has been squibbed and runs up to about the level of the quarry floor. From this point to within 30 ft. from the top of the quarry the hole is loaded with 30 per cent dynamite. The remainder is tamped only, no powder being needed, since the explosion will shear off the top 30 ft. From 400 to 800 lb. of dynamite is put into each hole, and is then detonated by connecting the electric exploders to a 500-volt circuit from the power plant. The usual blast is from 6 to 14 holes. In some blasts we have broken down 60,000 tons of stone at one time.

TABLE 1 SIZES OF ROCK GRIPPED AND PASSED BY 6 FT. BY 7 FT. ROLLS FOR DIFFERENT SPACING OF ROLLS

Center to Center of Rolls		Bottom to Bottom of Corrugations	Size of Maximum Gripped	Maximum Cube Gripped	Maximum Cube Gripped	Maximum Stone Gripped at 165 Lb. per Cu. Ft.	Ratio of Cubes Gripped	Maximum Cube through Rolls	Maximum Cube through Rolls	Maximum Stone through Rolls at 165 Lb. per Cu. Ft.	Crushing Ratio
Ft.	In.	In.	In.	Cu. In.	Cu. Ft.	Tons		Cu. In.	Cu. Ft.	Tons	
6	2	6	24	13824	8.00	0.66	1.00	75	0.043	0.0035	38.0
6	5	9	27	19682	11.40	0.94	1.42	262	0.15	0.012	16.3
6	8	12	30	27000	17.80	1.46	1.95	614	0.36	0.030	9.2
6	11	15	33	35937	20.80	1.72	2.60	1331	0.77	0.064	5.6
7	2	18	36	46656	27.00	2.22	3.35	2744	1.58	0.130	3.5
7	5	21	39	59319	34.20	2.82	4.28	4913	2.85	0.235	2.5
7	8	24	42	74088	42.75	3.53	5.35	8000	4.63	0.383	1.9
7	11	27	45	91125	52.50	4.33	6.60	12167	7.00	0.580	1.6
8	2	30	48	110592	64.00	5.28	8.00	17576	10.15	0.840	1.3
8	5	33	51	132651	76.50	6.31	9.60	24389	14.00	1.150	1.15
8	8	36	54	157464	91.00	7.50	11.30	32768	19.00	1.560	1.0

44 Steam shovels weighing 95 tons are used to load this stone into special steel skips for transportation to the crushing plant (Fig. 6.) In loading stones of this kind, it is necessary to handle them on the teeth of the steam shovel dipper and to roll them off into the skip. Steam shovel engineers become very adept in doing this, frequently loading a 5 or 8-ton stone in 20 seconds. The train of stone is then delivered to the foot of the incline at the crusher and is pulled up, three cars at a time, by an electric hoist. The skip is loose on the car and is dumped by another electric hoist, as shown in Fig. 3. The entire contents of the car slides into the hopper over the giant rolls.

Two distinct actions take place here: first, the sledging action due to the rapid blows (440 per minute) of the slugging knobs in striking the pieces of stone too large to be caught by the angle of grip of the rolls; second, the rolling action as the pieces are sledged off and caught between the rolls. It requires ordinarily from 5 to 20 seconds to reduce to 6-in. sizes a stone weighing 6 or 8 tons.

45 The delivery of energy is so great in crushing stones of such sizes in so short a time that the rolls at once slow down in speed. The writer has made a number of tachometer tests (Figs. 7 and 8) showing the action of these rolls while crushing a stone. To make these tests, two special Schaeffer and Budenberg recording tachometers constructed with their dials geared together, so they revolved in unison, were used. The recording arms of the tachometers were connected

TABLE 2 MINUTES LOST ON ROLL CRUSHER

	Giant	1st 36 in.	2d 36 in.	3d 36 in.
Belt	2585	72	203	426
Choked	....	383	532	802
Large stone caught in hopper	955	....	....	....
Gears	....	1872	300	....
Starting	44	.....	.....	.....
Bearings	168	282	.....	20
Bolts	.....	95	.....	317
Plates	699	91	.....	260
Shafts	337	.....	238	.....
Chain wobble	.....	417	634	.....
Sheared	.....	1361	150	518
Miscellaneous	311	187	.....	.....
Total minutes lost	5099	4760	2057	2343
Total hours lost	85	79	34	39
Per cent ran	98½	98¼	99¾	99½

one to each roll-shaft and the dials were revolved while the rolls were crushing the rock. By this means the speed of one roll with relation to the other could be determined at any instant. The tachometer chart (Fig. 7) shows that the slugger roll dropped in speed from 222 r.p.m. to 135 r.p.m. while crushing an 8-ton stone, and the regular roll (Fig. 8) dropped from 220 r.p.m. to 150 r.p. n., when they slowly regained speed until they reached normal.

46 While the second rock, almost as large as the first, was being crushed, the drop in speed was very much less. This may have been due to the fact that the stone was already partly shattered by dyna-

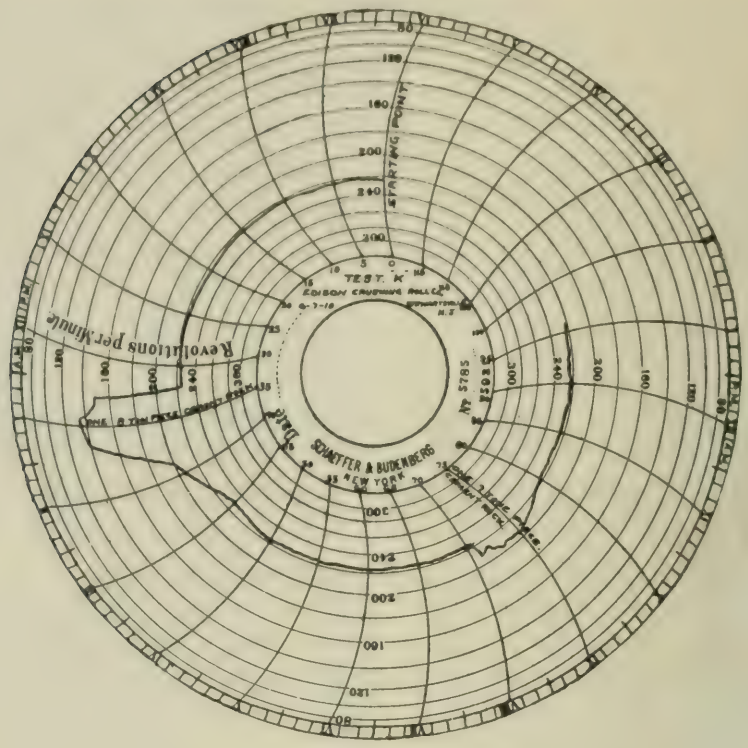

FIG. 7 TACHOMETER CHART FOR SLUGGER ROLL

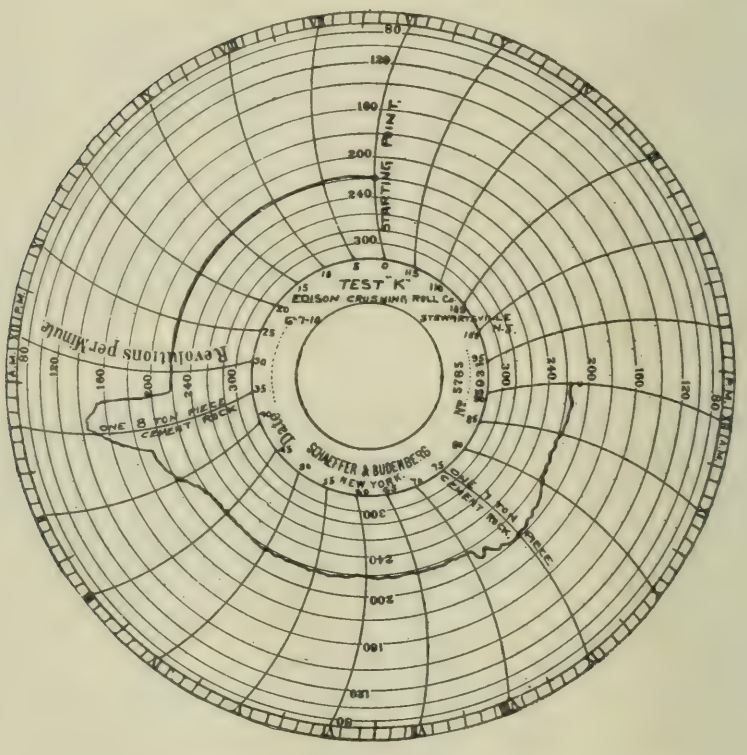

FIG. 8 TACHOMETER CHART FOR REGULAR ROLL

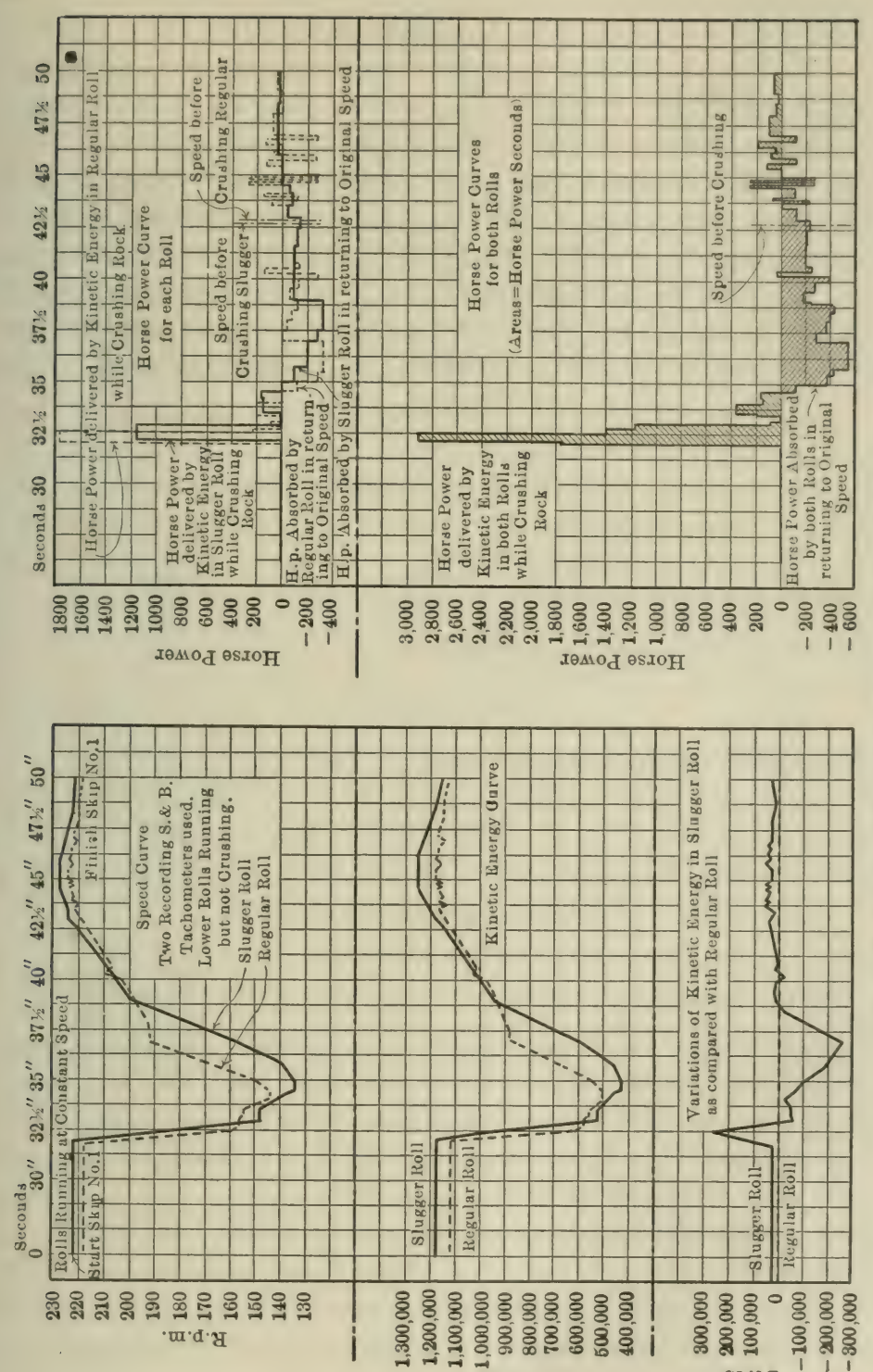
Seconds
0 30" 32½" 35" 37½" 40" 42½" 45" 47½" 50"
Rolls Running at Constant Speed
Start Skip No. 1
Finish Skip No. 1
Speed Curve
Two Recording S. & B. Tachometers used.
Lower Rolls Running but not Crushing.
Slugger Roll
Regular Roll
R.p.m.
230 220 210 200 190 180 170 160 150 140 130
Slugger Roll
Regular Roll
Kinetic Energy Curve
Ft.-lb.
1,300,000 1,200,000 1,100,000 1,000,000 900,000 800,000 700,000 600,000 500,000 400,000
Variations of Kinetic Energy in Slugger Roll as compared with Regular Roll
Slugger Roll
Regular Roll
Ft.-lb.
300,000 200,000 100,000 0 − 100,000 − 200,000 − 300,000
Seconds 30 32½ 35 37½ 40 42½ 45 47½ 50
Horse Power delivered by Kinetic Energy in Regular Roll while Crushing Rock
Horse Power Curve for each Roll
Horse Power delivered by Kinetic Energy in Slugger Roll while Crushing Rock
Speed before Crushing Regular
Speed before Crushing Slugger
H.p. Absorbed by Regular Roll in returning to Original Speed
H.p. Absorbed by Slugger Roll in returning to Original Speed
Horse Power
1800 1600 1400 1200 1000 800 600 400 200 0 − 200 − 400
Horse Power Curves for both Rolls
(Areas = Horse Power Seconds)
Horse Power delivered by Kinetic Energy in both Rolls while Crushing Rock
Speed before Crushing
Horse Power Absorbed by both Rolls in returning to Original Speed
Horse Power
3,000 2,800 2,600 2,400 2,200 2,000 1,800 1,600 1,400 1,200 1,000 800 600 400 200 0 − 200 − 400 − 600

FIG. 9 CURVES PLOTTED FROM TACHOMETER CHARTS

mite, or to the manner in which it struck the roll. In the latter case the slugger roll ran constantly at a higher speed than the regular roll.

47 Fig. 9 is a chart plotted from the dial records on the first stone crushed, showing the reduction in speed of the two rolls at any instant. From this chart was plotted the kinetic energy curve for each roll, the weight of each being about 25 tons. The variation in kinetic energy is also shown for the slugger roll as compared with the regular roll during this crushing operation. From these curves were plotted the horsepower curve, showing the horsepower delivered by the kinetic energy of each roll while crushing this stone, and also the horsepower absorbed by the rolls in returning to the original speed. In the

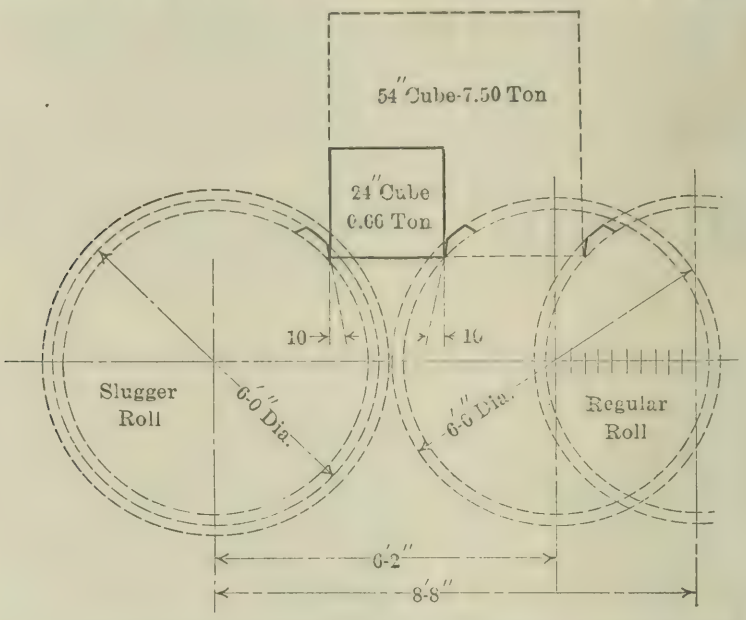

FIG. 10 DIAGRAM OF CRUSHING ROLLS SHOWING ANGLE OF GRIP

last diagram on the sheet, by combining the above curves, the horsepower delivered by the kinetic energy of both rolls while crushing rock is shown, also that absorbed by both rolls while returning to their original speed. This shows that for approximately half a second the two rolls delivered 2900 h.p. in kinetic energy, and the engine supplied about 600 h.p. additional, making a total of 3500 h.p. to crush the stone.

48 It can readily be seen how necessary it is to use the kinetic energy of the rolls when crushing rock, as the rated horsepower of the engine which drives these rolls and the three sets of smaller rolls is only 500 h.p. It also shows that practically all the stone was crushed within 3 seconds, while it required about 10 seconds for the rolls to regain their original speed.

49 This reduction in the speed of the rolls occurs through the slipping of the drive belt over the pulleys. The engine slows down somewhat, but only a small amount compared to the slippage. As this slippage occurs only for a short time it causes no serious trouble.

50 The slugger roll does the most work in crushing the stone and should show uniformly a greater reduction in speed than the regular roll. This, however, is not the case, since the action of the stone in passing between the rolls tends to gear the rolls together or to absorb energy from the faster roll and to deliver it to the slower roll.

51 The angle of grip of the rolls is not a definite angle and varies with the fracture of the stone and with the diameter and spacing of the rolls. Fig. 10 and Table 1 show these variations quite clearly.

52 The Edison Portland Cement Company has kept an accurate Time Lost account with each piece of machinery. An accurate statement of the total number of minutes lost on each roll and the causes of such stoppage for the years 1909 and 1910 are given in Table 2. The total loss of time is 237 hours, out of a possible running time of 4814 hours. The total delays due to the 4 rolls, therefore, amounted to 4.9 per cent of the possible running time, and the average tons crushed per operating hour to 224.

53 The actual cost of all material bought or manufactured by our shops for repairs on the rolls is available, but the actual labor of making the repairs cannot be accurately determined, since it is included in the item of repairs to the crusher, car hoist, dryers, conveyors, etc. The charges for material are itemized as follows:

Roll plates	$4454.90
Bearings	92.04
Gears, shafts, etc	732.78
Plate and coupling bolts	240.88
Hopper plates	242.88
Belts	2192.19
Miscellaneous	762.72
	$8718.39

54 During this time (two years) the plant reduced 1,024,409 tons of stone from 10-ton pieces, so that all would pass a $\frac{3}{4}$-in. screen. This makes a plate cost per ton of $0.0043; belt cost per ton of $0.0021; general repairs per ton of $0.0021 and total repairs per ton crushed of $0.0085.

55 All of these rolls are driven by a prime mover inadequate to start them from a state of rest, and it is necessary to use levers or some similar device in starting.

56 The strains set up when a 15-ton stone drops 10 ft. or more to the rolls are enormous, as are also the crushing strains, but they are largely taken up internally because the rolls act as an anvil. Only a comparatively small part of the shock is transmitted to the bearings or driving power.

57 The wearing surface of all the rolls is made of chilled cast-iron plates. On the giant rolls the chill is about 2 in. deep, while on the smaller rolls it is from $\frac{3}{4}$ in. to 1 in. deep. We have found that chilled iron, when properly made, wears much longer than manganese steel, and of course is much cheaper.

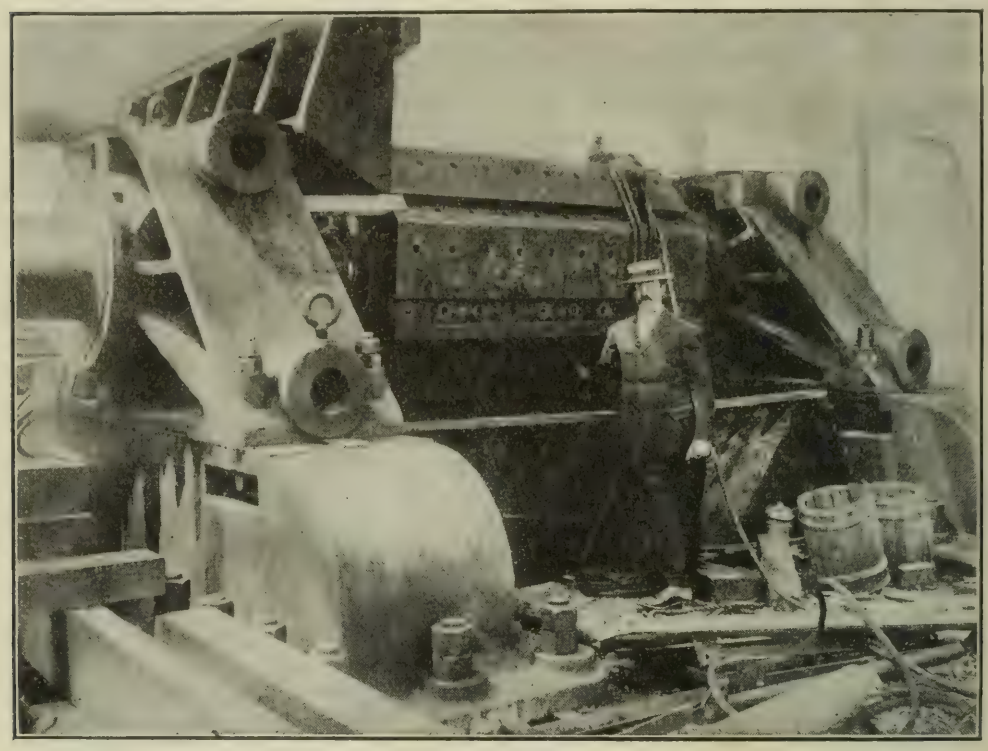

FIG. 11 GIANT ROLLS DURING ERECTION AT THE PLANT OF THE U. S. CRUSHED STONE COMPANY, CHICAGO

58 The capacity of the giant rolls is almost unlimited. In a recent test made on a pair of Edison rolls 6 ft. in diameter and 7 ft. long, 35 tons of stone were crushed in 32 seconds, or at the rate of 4000 tons per hr. This was done at one of the quarries of the Kelly Island Lime and Transport Company on a dolomite "run of quarry" loaded by steam shovels, the pieces being 4 to 5 tons or less. Side-dump cars were used on this test and they were arranged to dump automatically into the hopper over the rolls as the train was pulled by. It is

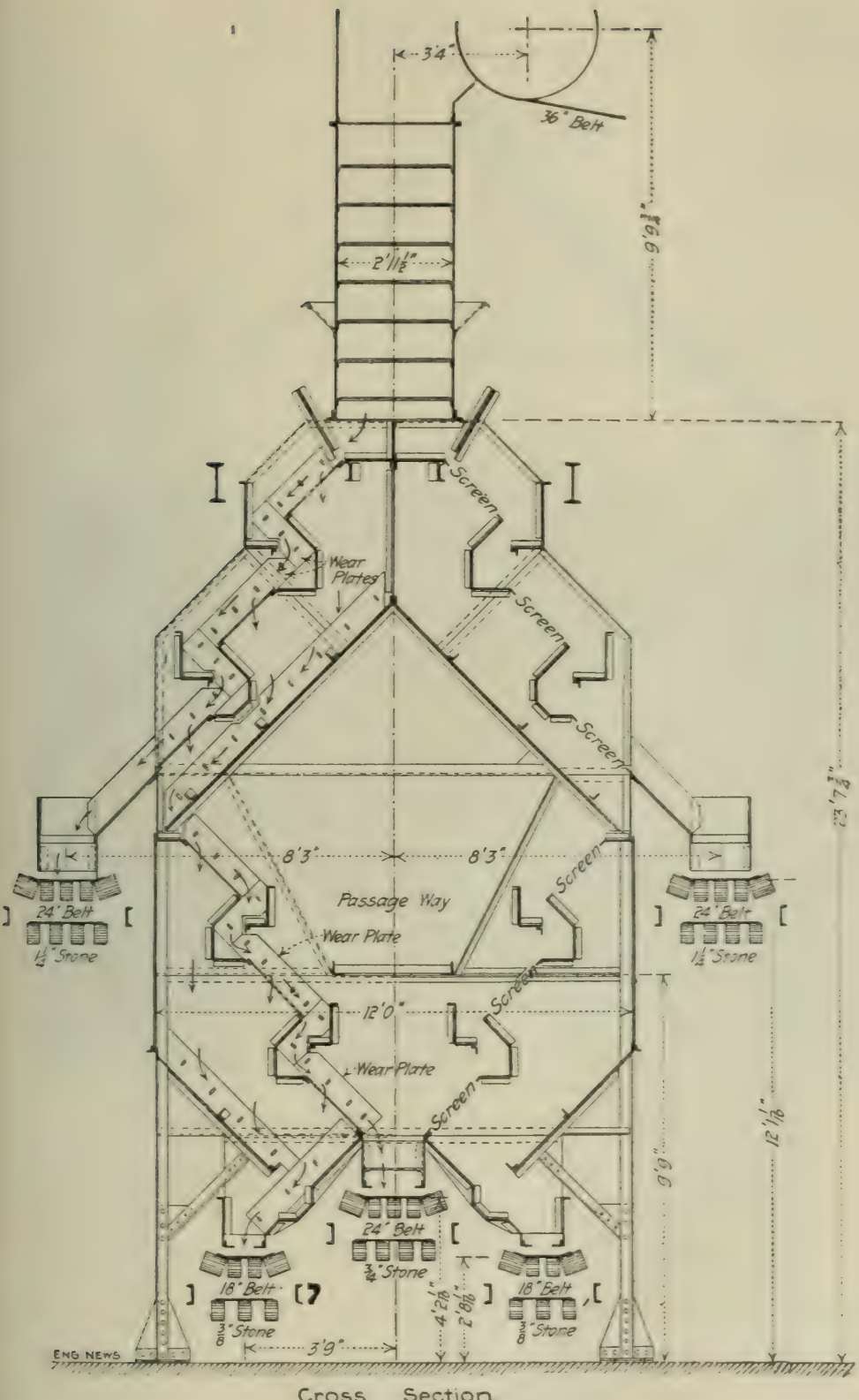

FIG. 12 SECTIONAL VIEW OF EDISON SIZING SCREEN

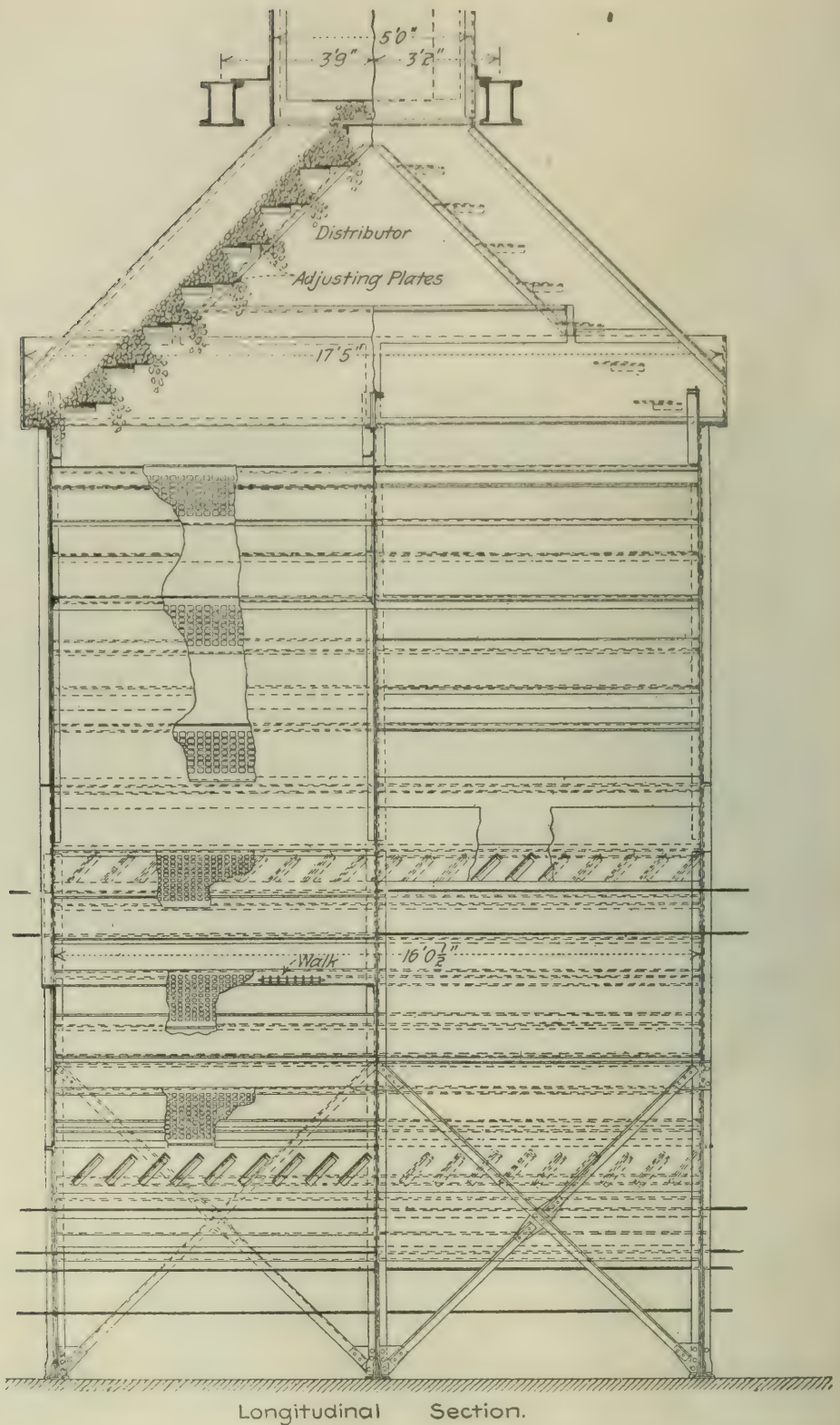

FIG. 13 END ELEVATION OF EDISON SIZING SCREEN

needless to say that at the end of the test the pan conveyor under the rolls was disabled and the rolls were running at a considerable reduction in speed.

59 The above figures may seem startling, but the theoretical capacity of the rolls is much greater when calculated on the same basis as that on which the capacity of smaller rolls is figured, and which the writer has frequently proved by actual tests to be correct.

60 The rolls are 6 ft. in diameter and 7 ft. long, are run at 185 r.p.m., and the average opening between the rolls is 9 in. This gives a surface speed to the rolls of 3487 ft. per min. or $\frac{3487 \times 7}{12} =$

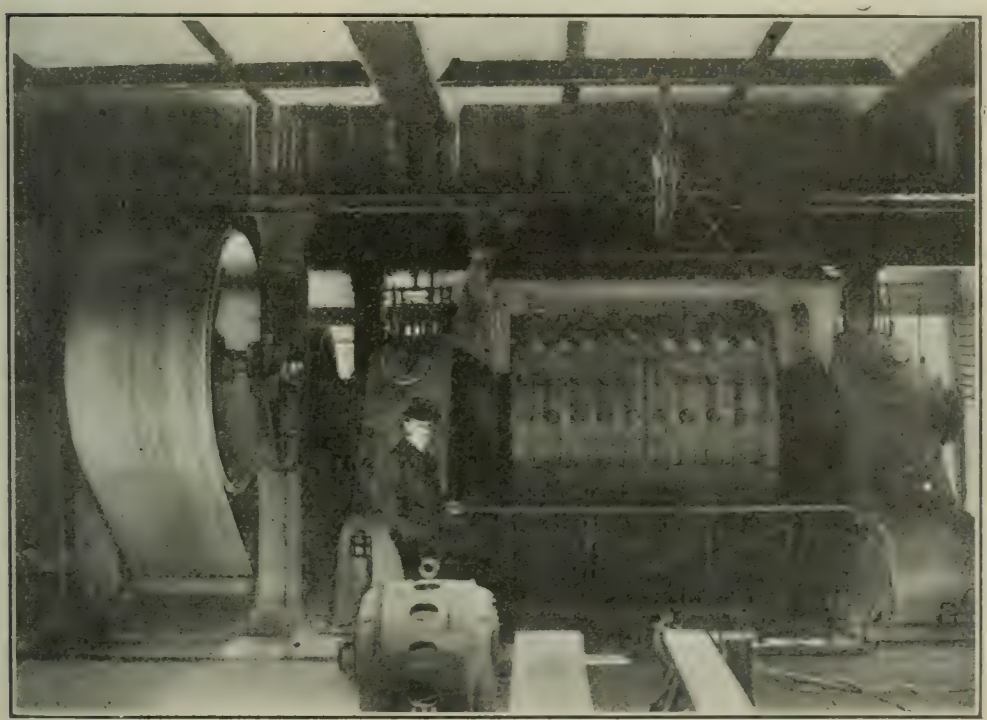

FIG. 14 GIANT ROLLS, 6 FT. BY 7 FT., AT TOMKINS COVE

18,306 cu. ft. passed per minute. Assuming 20 cu. ft. per ton, this would be equivalent to 915 tons per min., or about 55,000 tons per hr.

61 The average horsepower required to drive these rolls while crushing 3000 or 4000 tons per day is quite small, ranging between 100 and 150 h.p., but the momentary peak loads are very much greater. One of the companies operating a pair of these rolls 6 ft. by 7 ft., motor driven, buys its power with certain peak load specifications. In order to reduce the peaks it has set a circuit breaker to cut off the current at about the normal load of the motor. After the stone is

crushed the current is turned on the motor, and the roll again brought up to speed. It frequently happens that the current is cut off the motors by this method as many as 50 times a day.

62 Fig. 11 shows a set of 6 ft. by 7 ft. rolls being erected at the plant of the U. S. Crushed Stone Company, Chicago, Ill., and gives an idea of the heavy construction necessary to withstand the great shocks. This also shows the mandrels to which the plates are attached.

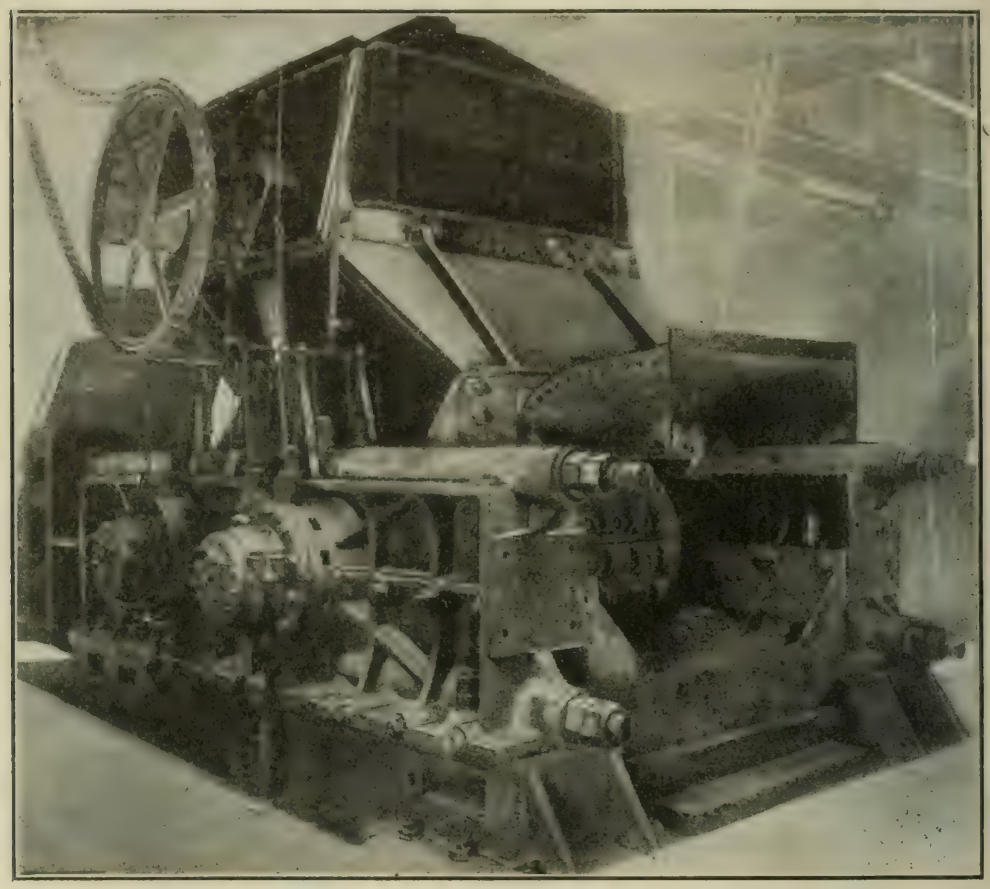

FIG. 15 INTERMEDIATE ROLLS AT TOMKINS COVE—4 FT BY 4 FT.

63 The crushing plant of the Tomkins Cove Stone Company, Tomkins Cove, N. Y.,[1] which was put in operation last fall, has, as far as the writer knows, the largest capacity of any plant in the world. Its extreme simplicity is one of the most striking features, and its design throughout is for a capacity of 1000 tons per hour. The stone,

[1] This plant was more fully described and illustrated in Engineering News, January 12, 1911.

after being loaded with steam shovels in the quarry, is dumped into a pair of 6 ft. by 7 ft. rolls, reducing it approximately to 8-in. sizes. Under these rolls there is a hopper having a capacity of about 30 tons. The stone is fed from this hopper by feed rolls to a set of 4 ft. by 4 ft. rolls, which run 250 r.p.m. This reduces the stone to about $3\frac{1}{2}$-in. sizes when it goes directly to a set of 4 ft. by 3 ft. rolls and is reduced to about $1\frac{1}{2}$ in. A large pan conveyor receives the stone and lifts it to an Edison stationary screen, which returns anything over $1\frac{1}{2}$-in. sizes to the lower rolls for re-crushing. The product from this

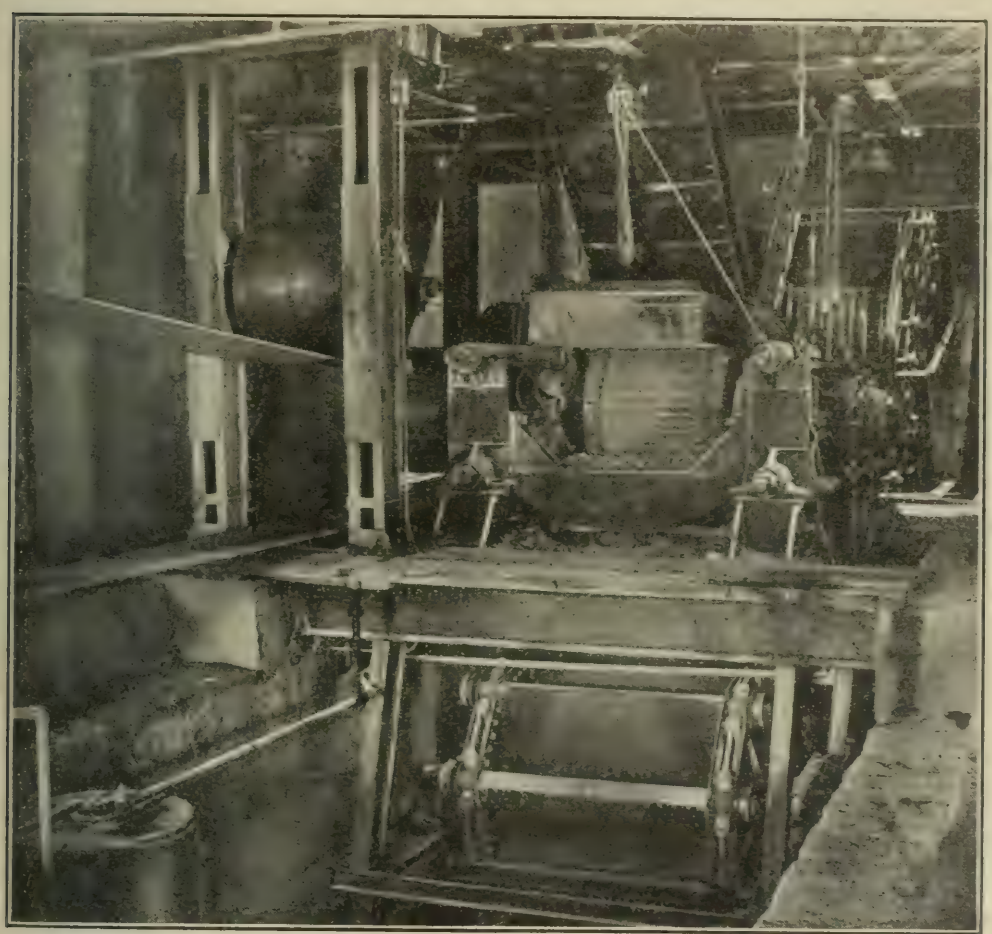

FIG. 16 FINAL ROLLS AT TOMKINS COVE—4 FT. BY 3 FT.

screen is carried by a 36-in. belt conveyor to the Edison sizing screens (Figs. 12 and 13). Here it is divided into three sizes, known commercially as $1\frac{1}{4}$-in. stone, $\frac{3}{4}$-in. stone and $\frac{3}{8}$-in. stone. It is carried by belt conveyors to concrete bins having a total capacity of 20,000 tons, from which it can be withdrawn and delivered by belt conveyors directly to barges or railroad cars.

64 The size of stone may be varied as desired by changing the openings between the rolls or the size of the openings on the screen plates.

65 Figs. 14, 15, 16 show the giant, intermediate and final rolls at the Tomkins Cove plant.

No. 1311 c

POWER AND HEAT DISTRIBUTION IN CEMENT MILLS

By L. L. Griffiths, Cement, Dallas Co., Tex.

Member of the Society

66 This paper considers the methods employed in cement mills for converting the dormant thermal power of fuel into active and productive mechanical motion. It has been the good fortune of the writer to have been in direct charge of several manufacturing plants and data are given with reference to five cement plants which use the thermal power of fuel in different ways.

67 The primary means of developing this thermal power of a fuel is by means of the application of the heat evolved, either by total or partial combustion. If by total combustion, the first means commonly employed for the first stage of the power transmission is steam, either saturated or superheated. The prime mover actuated by this steam, whether it be a reciprocating engine or a turbine, is the second step of the power transmission. Then we have the further transmission of this power by means of the belted, geared or direct connection of this prime mover to a lineshaft, thence by means of further belts, gears, ropes or chains from the lineshaft to the machine finally operated. Or we have the prime mover actuating a generator from which the power in changed form is transmitted through wires to motors operating the machines, either by direct connection or by means of lineshafting, belts, gears, ropes or chains.

68 Where the first stage is the partial combustion of the fuel resulting in the production of an explosive gas, the prime mover is an internal-combustion engine connected similarly to that just described.

69 In considering the several plants as set forth in Table 3, a brief description is given both of the power installation and the mill

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Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

arrangement, since there is a considerable difference of opinion as to which form of power consumption is the most economical type for crushing, grinding and pulverizing machinery.

70 The item of fuel herein considered as the source of power, is roughly 30 per cent of the manufacturing cost of a barrel of cement

TABLE 3 DATA UPON THE POWER DISTRIBUTION IN FIVE CEMENT PLANTS

	No. 1	No. 2	No. 3	No. 4	No. 5
Nominal Capacity, Bbl	1000	1200	2000	2000	2500
Actual Capacity, Bbl	960	1220	2180	1176	1800
Indicated Horse Power					
Gross Developed	805	1020	1694	1480	2430
Per Bbl. of Cement	0.839	0.835	0.777	1.26	1.35
Hours, per Bbl	20.15	20.04	18.65	30.75	32.4
Coal Cost, per Ton	$2.76	$2.12	$2.45	$1.25	$1.50
Coal Cost, Cents per H.p-hr	0.00655	0.00325	.00411	0.00419	0.0019
Lb. per H.p.	5.32	3.44	3.8	6.7	2.87
Proportion to Departments					
B_1, H.p	.....	.....	.....	189	136
%	.....	.....	.....	12.7	5.6
B_2, H.p	.....	.....	.....	547	724
%	.....	.....	.....	36.8	29.8
B, H.p	387	416	602	736	860
%	48	40.75	35.5	49.5	35.4
C, H.p	38	37	159	86	170
%	4.5	3.5	9.5	5.8	7.0
D_1, H.p	.....	.....	.....	.....	.....
%	.....	.....	.....	.....	.....
D_2, H.p	.....	.....	.....	.....	.....
%	.....	.....	.....	.....	.....
D, H.p	300	476	750	531	962
%	38	46.75	44.5	35.7	39.6
Coal, Grinding	80	90	183	132	.....
%	9.5	9.0	10.75	8.86	.....
Power for Auxiliaries	.....	.....	.....	.....	438
%	.....	.....	.....	.....	18

and each barrel of cement manufactured represents the conversion of from 2,000,000 to 3,000,000 B.t.u. Of this total amount, 25 per cent is used in the generation of power in the actual mechanical operation of the machinery of the plant, so that it may readily be seen that the means of converting and transmitting this power is worthy of study and consideration.

PLANT NO. 1, 1000-BBL. NOMINAL CAPACITY

71 When operating 100 per cent of the machinery 100 per cent of the time, the daily output of plant No. 1 would be 1000 bbl. of Portland cement. Briefly, its general arrangement is as follows: Stone from the quarry in man-size pieces is crushed in a gyratory crusher, dried in a rotary drier, further reduced in rolls and finished pulverized in Fuller mills. The raw material is burned in rotary kilns, and the cement clinker prepared for pulverizing by passing through toothed crushers, and pulverized in 30-in. Griffin mills.

72 The boiler equipment of the power plant consists of three horizontal tubular boilers of 75 h.p. each, four horizontal tubular boilers of 100 h.p. each and one B. & W. water tubular boiler of 340 h.p. The ash-pits of all the boilers are equipped with McClave steam blowers with steam pressure at 100 lb. They are using West Virginia bituminous slack or run of mine, but its use is uneconomical because the grates are set for hard coal. About 60 per cent of the raw end of this plant is driven by an 18 in. by 40 in. Wright non-condensing engine and the balance, or 40 per cent, of the raw end, and 100 per cent of the finishing end, is driven by a 20 in. and 34 in. by 42 in. tandem compound Corliss engine operating at 100 r.p.m. in connection with a jet condenser and cooling-tower installation. The kiln room is operated by one 11 in. by 11 in. Buckeye non-condensing 150-r.p.m. engine. With this power installation the plant turns out a barrel of cement for 0.839 h.p.

PLANT NO. 2, 1200-BBL. NOMINAL CAPACITY

73 Plant No. 2 of 1200-bbl. nominal capacity, operates at an output better than 100 per cent because of the excellence of the operating force. The general arrangement is as follows: Stone from the quarry is crushed in gyratory crushers of the same make and size as in plant No. 1, dried in a rotary drier, also of the same make and size as in plant No. 1, but the material is now ground in ball mills and is finished pulverized in tube mills, then burned in kilns of the same size as those of plant No. 1. The cement is prepared for pulverizing and is pulverized as in plant No. 1.

74 The boiler equipment of the power plant consists of eight double-drum elephant boilers each of 150 h.p. capacity. The furnaces are equipped with balanced draft, burning Buckwheat anthracite coal. The raw end of this plant is driven by one 20 in. and 36 in. by 42 in. cross-compound condensing Corliss engine operating at

87 r.p.m. The finishing end is driven by a duplicate of this engine, and the kiln room is operated by one 11 in. by 18 in. 150-r.p.m. Buckeye non-condensing engine. The coal house, with its handling, drying, crushing and pulverizing of coal for kiln fuel, is operated by one 11 in. by 10 in. vertical 230-r.p.m. Westinghouse steam engine, and one 20 in. by 16 in. 250-r.p.m. Westinghouse engine. This plant produces a barrel of cement on 0.835 i.h.p.

PLANT NO. 3, 2000-BBL. NOMINAL CAPACITY

75 Plant No. 3 was more recently constructed than either No. 1 or No. 2 and is laid out in a somewhat better manner. This fact together with a more efficient working force, results in an output better than the rated output of 100 per cent. The crushing, grinding and pulverizing machinery of the raw ends, kilns and auxiliaries of the burning department, and crushers and pulverizers of the finishing end, were the same as in plant No. 2, a greater number of the same size units being used for the increased capacity.

76 The boiler equipment of the power plant consists of seven 350-h.p. B. & W. boilers fired by Westinghouse stokers using anthracite pea coal, both coal and ashes being automatically handled by conveyors and elevators. The boilers are also equipped with superheaters and the raw end is operated by one 20 in. and 40 in. by 54 in. cross-compound Corliss condensing engine of 74 r.p.m. The finishing end is operated by a duplicate of this engine, and the kiln room by one 16 in. and 28 in. by 42 in. tandem compound Corliss condensing engine at 85 r.p.m. The coal house is operated by one 12 in. and 24 in. by 42 in. tandem compound Corliss condensing engine of 80 r.p.m. This plant produces a barrel of cement on 0.777 i.h.p. The prime movers in every instance are belted to a main lineshaft, the individual machines being in turn driven from this lineshaft by belt, chain, rope or gear drives.

77 These three plants are known in the cement business as direct-driven plants: that is, in such plants the prime mover is belted to a lineshaft and the individual machines are in turn driven by belts, gears, chains, etc., from this lineshaft. Plants Nos. 4 and 5 are known as electrically-driven plants.

PLANT NO. 4, 2000-BBL. NOMINAL CAPACITY

78 Plant No. 4 with a nominal capacity of 2000 bbl., although a comparatively new plant, has never been operated at better than 60 per cent of its nominal rating, due to a very poor arrangement

of mixing, conveying and pulverizing machinery. On account of the load condition, however, the electrical efficiency is high, the motors operating at from 100 to 125 per cent full load; and the maximum efficiency of the induction motor is at about 120 per cent rated load.

79 The general arrangement of this plant is as follows: Stone in man-size pieces is reduced by passing it through two gyratory crushers; the shale is reduced by passage through toothed rolls. It is dried in a rotary drier, mixed with limestone and further reduced with it in a Williams mill. The material is then passed through two rotary mix driers to 30-in. Griffin mills used as preliminaries to tube mills, in which the raw material is finished. This is burned in kilns of the same size as those in the No. 2 and No. 3 plants. The clinker is prepared by passing through a toothed crusher and rolls, and pulverized in 30-in. Griffin mills.

80 The boiler equipment of the power plant consists of two 250-h.p. Cahall vertical boilers, four 250-h.p. O'Brien–Heine boilers, and one 375-h.p. O'Brien-Heine boiler. The fuel used is Southern Indiana bituminous coal and with the exception of the Cahall boilers, the setting and grates are for hard coal. The furnace efficiency is therefore low. The prime movers in this plant are two vertical Curtis turbines, one direct-connected to a 1000-kw. 3-phase, 25-cycle, 440-volt alternator, and the other to an 800-kw. alternator. Each individual machine throughout the plant is driven by an individual motor of proper size by a belt from the motor pulley to a shaft and from the shaft to the machine either by gears, belt, or chain drive. This plant consumed 1.26 h.p. per bbl. of cement produced.

PLANT NO. 5, 2500-BBL. NOMINAL CAPACITY

81 Plant No. 5 comes under the same heading as plant No. 4, but is unique in that it is the only one which starts with partial combustion of the fuel, and discarding steam as a means of power transmission between the fuel bed and the prime mover, uses the gas direct from the fuel in an internal-combustion or gas engine. Due to the difficulty of mixing and combining the raw materials of the section in which the plant is located it is operating only at about 80 per cent of its capacity at the output figures given. The basis of comparison therefore, for the grinding, pulverizing and burning machinery, which is operating on cement rock in the Lehigh Valley, is not a fair one.

82 The stone is loaded with a steam shovel, crushed in a gyratory crusher, further reduced in a Williams mill, and dried in rotary driers. The shale or clay is loaded with a steam shovel, run through a dry pan mill, dried, and meets the stone at the weigh bin where the mixture is proportioned. It then passes through a rotary mixer to ball mills and is finished in tube mills, burned in rotary kilns, and the cement clinker then crushed in ball mills and finished in tube mills.

83 Electrically, the conditions are practically the same as those described for plant No. 4.

84 The power plant consists of six up-draft producers in place of the usual steam boilers, the generating gas operating three 34 in. by 48 in. horizontal tandem gas engines driving three 750-kw. 3-phase, 25-cycle, 550-volt alternators.

85 It may appear that this paper emphasizes the relative efficiencies of different raw materials, types of grinding machinery and engineering arrangements. The presence of these as variables is admitted, especially when the plants cited are in different localities, but fortunately we have exactly the same types of machinery and in most cases the same size and make in the comparisons given, if not throughout the entire plant at least throughout entire departments which represent the several stages in the manufacture of cement from the raw materials.

86 In explanation of the lettering of the different departments in Table 3 the quarry is considered as A; preliminary crushing, secondary breaking down and storing of the raw materials as B_1; the drying, grinding and pulverizing of the raw material as B_2; subdivisions B_1 and B_2 as the raw end B; burning the raw material to cement clinker and cooling it as department C; crushing, grinding and pulverizing the cement clinker as department D, D_1 being the preliminary crushing and grinding, and D_2 the finishing or pulverizing.

87 Comparing the figures in Tables 3 and 4 in connection with the brief description of the plants, it is seen that a direct-driven plant shows a power economy greater than an electrically-driven plant.

88 Plants Nos. 2, 3 and 5 are comparable in departments B_1 and B_2 with but small opportunity of error in that the size, type and make of the granulating and pulverizing machinery are identical. It will be noted that a barrel of finished raw material at the kiln bins, department B, consumes as follows:

	I.H.P. Total	I.H.P. Per Bbl.
Plant No. 1	387	0.403
Plant No. 2	416	0.341
Plant No. 3	602	0.276
Plant No. 4	736	0.626
Plant No. 5	860	0.478

89 Plants Nos. 3 and 4 have twice the number of kilns, feeders for both pulverized coal and raw material, low-pressure air systems, elevators and auxiliaries that plant No. 2 has. Plant No. 5 has the same number of kilns, etc., as plant No. 2, but the kilns have twice the area and length and the auxiliaries are twice as large. We have as a comparison, however:

	I.H.P. Total	I.H.P. Per Bbl.
Plant No. 1	38	0.0396
Plant No. 2	37	0.0303
Plant No. 3	159	0.073
Plant No. 4	86	0.0732
Plant No. 5	170	0.0945

90 In the clinker-grinding and finishing departments *D*, we find a greater difference in the machinery used, that is, when comparing plants Nos. 2, 3 and 5; but the department *D* of plant No. 4 is comparable with that of plants No. 2 and No. 3 with reference to the grinding machinery used, and we have:

	I.H.P. Total	I.H.P. Per Bbl.
Plant No. 1	300	0.312
Plant No. 2	476	0.390
Plant No. 3	750	0.341
Plant No. 4	531	0.452
Plant No. 5	962	0.534

From the above it will be noted that in practically every instance the direct-driven plant is more economical in power consumption, than the electrically-driven plant.

91 In considering this subject, it has appeared to the author that the pounds of coal used per indicated horsepower do not show the true condition, so the equivalents in British thermal units are given in Table 4.

TABLE 4 HEAT UNITS PER I. H. P. PER BBL. CEMENT

Plant No.	Coal, Lb. per i.h.p-hr.	B.t.u. per i.h.p-hr.	Power Coal, Lb. per bbl. Cement	B.t.u. per bbl. Cement
1	5.32	71,022	107.19	1,430,960
2	3.44	40,560	68.94	792,782
3	3.80	45,600	70.87	850,440
4	6.7	82,745	200.03	2,470,305
5	2.87	21,094	92.99	683,462

92 From Table 4 it appears that the power plant No. 5, using the most direct method of transmitting the thermal power of the fuel to the prime mover, i.e., by means of the gas generated by partial combustion of the fuel in a producer, shows the lowest B.t.u. consumption and highest thermal efficiency, but on account of the losses inherent in the balance of the power transmission system, uses more i.h.p. per bbl. of cement produced than do plants Nos. 1, 2, 3 and 4. The same is true of plant No. 4 as compared with plants Nos. 1, 2 and 3, from which it appears to the author that in addition to all the losses inherent in a direct-driven plant, the electrically-driven plant has in addition those decreased efficiencies representing losses in the conversion of mechanical power to electrical power and back from electrical power to mechanical.

93 We have, for instance, in plants Nos. 1, 2 and 3 the fuel fired under boilers, generating steam which in turn is used in the engines giving motion and power to the lineshaft. Through the aid of a belt this power is then transmitted to the several machines by means of belts, gears, chains, ropes, etc. In this arrangement we have the losses shown in column 1.

94 In plant No. 4 the fuel is fired under boilers, and steam is used in the turbine to generate electrical current in the alternators, the current produces motion and power in the motors which is transmitted to the machines, through a belt to a countershaft, and thence by belt, gearing, chain or rope to the machine, and the losses are as shown in column 2.

95 In plant No. 5 the fuel is fired into a producer which supplies gas to an engine operating an alternator generating current from which point the cycle is the same as in plant No. 4, and the losses are as shown in column 3 which follows:

Nos. 1, 2 and 3	No. 4	No. 5
Furnace	Furnace	Producer
Boiler	Boiler	Engine (gas)
Engine	Engine (turbine)	Generator
Belting	Generator	Line
Lineshaft	Line (circuits)	Motor
Lineshaft	Motor	Belt
Belting	Belt	Lineshaft
	Lineshaft	Belt
	Belt	

96 The above are all comparable on a monetary basis, but time and space will not permit going into this. It is appreciated that, given a water-power proposition, the proper installation is a hydroelectric one, but it is believed that the most economical cement plants of the future, that is, from a power consumption standpoint, will be developed along the lines of a producer as the first step in the cycle of power conversion and transmission; then an internal-combustion engine operating the main departments direct-connected to the lineshaft, and the waste heat from these engines utilized in an exhaust or low-pressure steam turbine operating direct-current generators, which current so generated will be used for lighting and the operation of isolated and remote machines.

No. 1311 *d*

DISCUSSION ON CEMENT MANUFACTURE

H. Struckmann (written). In Par. 26, Professor Landis calls attention to the value of using the heat from the discharged clinker, stating that this practice has proved unsatisfactory in this country. The writer wants to add his testimony that utilizing the heat in the discharged clinker is of the greatest value for preheating the air necessary for combustion. In Europe when the first rotary kilns were installed, the heat was pulled by exhausters from the underlying rotary cooler and blown into the kiln mounted above it. Considerable difficulty was experienced in handling the large volume of air at the temperature at which it was received from the cooler.

The modern installations are now practically all using a process by which fans are blowing the necessary amount of cold air into the cooler, which is virtually put under pressure. This air, after having absorbed the heat from the clinker, is delivered through air ducts, entering the kiln on both sides of the hood, at a temperature as high as 1000 deg. fahr. The advantage of this installation is very evident, since the cold air handled by the fans is of very much smaller volume than the heated air. The heat available in the clinker is not only sufficient to furnish the necessary air for combustion, preheated to a temperature of 1000 deg. fahr., but also to dry the coal preparatory to the grinding process, resulting in a large saving of fuel.

The statements which Professor Landis makes about the wet process are not borne out by experience with the most modern wet installations in Europe. The explanation in Pars. 17 and 18 indicates that Professor Landis is comparing the old-style short-kiln installation, where the material was dried before it was ground, with a modern dry-process installation with long kilns. Such a comparison is hardly fair to the modern wet plants now being designed and operated in Europe. There is no question but that a 60-ft. kiln burning wet is

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Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

an extremely wasteful proposition, but with the development of the long kiln the situation has been entirely changed.

It has been conclusively proved that grinding the raw material in wet condition with proper machinery reduces the horsepower from 30 to 40 per cent, besides increasing the fineness very materially and entirely eliminating the mill dust problem, which deserves a good deal of attention.

If we combine the drying and burning in one rotary, we would have a kiln about 185 ft. long, basing the figures on a 60-ft. dryer and 125-ft. kiln, the two sizes now most generally used.

As a general rule in comparing the wet and dry processes, only the 125-ft. dry kiln is compared with the same size wet kiln, no attention being paid to the fact that the dry kiln requires an additional 60-ft. rotary to take care of the drying of the material. Of course as a result greater economy is apparently effected on the dry process. If, however, we compared the burning unit of the dry process with the corresponding burning unit of the wet process, the result would be entirely different.

The writer has made a number of experiments in order to ascertain how many feet of the kiln are required to evaporate the 30 per cent of water contained in the slurry, as it is being fed to the modern wet kiln, and finds that approximately 35 to 40 ft. is all that is necessary to put the material in virtually the same condition as when being fed to the dry kiln. If we deduct this 35 to 40 ft. from the wet 185-ft. kiln, we have a larger kiln with a resulting larger output and lower fuel consumption.

There are quite a few plants in this country today which would turn out a much more superior and reliable cement under the wet than the dry process, because in handling the slurry a perfect mixture is obtained. While this would not be the case with the Lehigh Valley plants, where a natural cement mixture is found, judgment ought scarcely to be passed on a process based on the condition of one small part of a country. There are a number of plants in the West in which the raw material in natural condition contains from 10 to 20 per cent of moisture, requiring a large amount of heat to evaporate, and considering all these facts, the writer fails to see why the wet processes should be eliminated entirely in the investigation of a cement proposition.

Professor Landis failed to touch on one subject in connection with the wet process, namely wet grinding, and the laboratory tests given in Table 5 may be of some interest. In carrying out these tests the

raw material was prepared to a fineness ranging between 20 and 30 mesh sieve and 1500 grams were used as a charge in a laboratory mill. Ten separate tests were run, each lasting 55 minutes. The first test was run with dry material and afterwards various percentages of water were added, as indicated in the test.

From these tests it is clearly demonstrated that from 30 to 32 per cent of water gives the greatest efficiency in grinding the material. With the increased fineness made possible by the wet grinding process, a great deal of fuel is saved. In some of the plants with which the writer is connected, natural gas is used for fuel and each kiln is equipped

TABLE 5 LABORATORY TESTS ON WET GRINDING

Test Number	Per Cent H_2O	100-Mesh	200-Mesh
1	0	92.0	75.0
2	25	98.4	80.8
3	28	99.8	87.0
4	30	99.2	86.8
5	32	99.6	84.8
6	35	96.8	80.6
7	37	96.8	....
8	40	96.6	80.2
9	45	96.4	80.8
10	50	95.2	79.6

with separate gas meters, giving a complete record of the amount of fuel used in each kiln. Experiments carried on for long periods with a varying fineness of the raw material showed an average of 56 cu. ft. of gas for each per cent of greater fineness between 88 and 96 per cent through the 100-mesh sieve. In other words, each percentage of finer raw material effected a saving of about 47,000 B.t.u. per bbl. of clinker produced. Besides this saving in fuel, it was found that the capacity of the kilns, which were 7 ft. 6 in. by 125 ft., was greatly increased, the output per 24 hours being as follows:

88.5 per cent fineness.....454 bbl. per day
90.0 per cent fineness.....534 bbl. per day
92.5 per cent fineness.....591 bbl. per day
94.0 per cent fineness.....600 bbl. per day
96.0 per cent fineness.....620 bbl. per day

In view of these facts it is a question whether the wet process as it is developed today does not represent the most scientific method of

producing Portland cement, where the raw materials are more or less mixed and contain a large percentage of moisture in natural condition.

P. C. VAN ZANDT (written). The crushing of rock is probably one of the most ancient human industries, and the course of its progress from a crude process into one worthy of scientific regard is interesting and instructive.

In Par. 42, Mr. Mason states that before Mr. Edison constructed his giant rolls the largest rolls in use were those known as Cornish rolls, which were geared together. These rolls were also built at that time for separate belt drives without being geared together, but, with only one or two exceptions for use on rock, the roll shells were smooth-

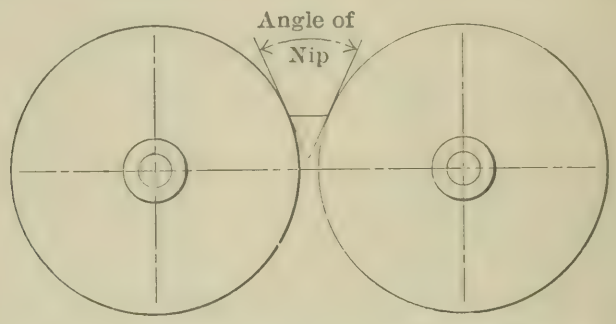


FIG. 17 ANGLE OF NIP ON SMOOTH ROLLS

faced. Rolls for crushing coal were then built as they are now with spikes or projections and, as far as the writer can remember, coal rolls were hardly ever built smooth-faced, nor rolls for use on rock other than smooth-faced.

In the crushing of rock the angle of nip as shown in Fig. 17 determined the largest piece of rock that could be fed to a set of smooth rolls, while no such limitation existed in the crushing of coal with spiked rolls, which automatically broke the large pieces of coal into fragments small enough to be nipped. Mr. Edison's rolls, like his rotary kiln, were unquestionably very much larger than anything in use at that time and he alone seems to have foreseen the direction of growth of the industry. The comparison of the kinetic energy stored in the rolls in use at that time and in the rolls built by him shows the magnitude of his step, while other builders of these machines were merely creeping along from one size to the next.

Mr. Mason's comparison of the maximum size of rock fed to a gyratory crusher and to a set of rolls is slightly unfair to the former

machine. The largest gyratory crusher built at the present time has two openings 48 in. wide by approximately 9 ft. long, this opening being on a curve somewhat the shape of the letter *C*. The writer has seen a piece of rock fed to a gyratory crusher of the 42-in. size, the dimensions of which were approximately 3 ft. by 5 ft. by 9 ft., this rock being dropped end down and going through the machine as though it were being eaten up. The writer is further of the opinion that the maximum sized piece fed to a set of 6 ft. by 7 ft. rolls would have to be more nearly 7 ft. in diameter than a 7-ft. cube and of spherical shape, to have it turned over by the rolls and new surfaces presented for crushing. However, in the writer's opinion such a set of 6 ft. by 7 ft. crushing rolls will take a larger piece of rock for crushing than any other machine in existence in the world today.

In Par. 44, Mr. Mason describes the jiggling onto the car or skip by the steam shovel operators of a piece of rock too large to go from the teeth of the dipper through the dipper itself. The writer has seen this done a great many times and judges that in some quarries it is rather difficult to prevent. The limit of size of rock that can be economically handled, and there must be a limit somewhere, is, in the writer's opinion, the biggest piece that can be handled *through* the dipper. The steam shovel operators like to make a showing for tonnage and efficiency in the quarry by sending up to the crusher as large pieces as they can handle. Occasionally a piece which is too big for the hopper of a set of rolls or a gyratory crusher is sent up, which has to be broken or blasted or returned, causing considerable delay at the crusher, which together with the expense represents a larger cost than the breaking of a dozen of these pieces in the quarry. Damage to the quarry cars or skips caused by dropping these big pieces of rock is also troublesome. The kinetic energy in the piece of rock striking the car compares favorably with that in the crushing rolls, when in one case the cars are smashed and in the other the rock. By comparison a large piece of rock handled *through* the dipper can be laid in the car very gently.

The writer has examined the tachometer record of test *K* very carefully and does not understand why these records which show the drop in speed of the two rolls so clearly, do not distinguish between the slugging and crushing action. It may be that Mr. Mason has other charts which do show this, and it would be interesting to have his opinion regarding the crushing action of these large rolls upon large pieces of rock. A piece, perhaps the size of a roll top desk, is dropped into the rolls and is first subjected to what may be called a slugging

action, in which it is tossed around in the hopper, the corners broken off, and finally shattered into pieces small enough to be caught within the angle of nip of the rolls. The second or crushing operation then takes place and the rock caught in the angle of nip is immediately carried through the rolls and crushed by pressure to a size sufficiently small to pass the available space between the rolls.

It would seem that the maximum tendency to vary the speed of one roll in relation to the other would exist in the slugging action, when first one and then the other roll strikes a large piece of rock; while in the crushing action, as Mr. Mason states in Par. 50, the action of the stone in passing between the rolls would tend to gear the rolls together, in other words, to cause them to rotate at the same speed. The curves plotted from the tachometer charts show an almost simultaneous drop in speed during the slugging action, with a difference in time of a small fraction of a second, while the recovery of speed in the two rolls, probably after the bulk of the crushing action has taken place, shows a much greater difference in speed, the curve giving apparently the reverse of what the writer would suppose the action to be.

In Par. 54, Mr. Mason gives the cost of repairs per ton of rock crushed, which the writer thinks is very good. He has seen records of lower costs of repairs on gyratory crushers, but as a partial offset, it must be remembered that they will not take pieces of rock so large. The horsepower required in the rolls per ton of rock crushed is also greater than the gyratory crushers, but the same partially offsetting advantage also applies here.

The questions raised in Professor Landis' article, Par. 12, namely, how much does it cost in investment, interest, repairs, depreciation, etc., in a rock-crushing plant to provide means for crushing the few larger pieces of rock that come to the crusher in the course of a day's operation, and does the saving affected more than offset the additional cost, must be answered for each individual case, depending upon the cost of installation, location, the desired capacity, the kind of rock and the use to which the rock is put when crushed. We must reach a limit, where Mr. Soper's law of pivotal points or decreasing returns[1] will apply, and in the writer's opinion, this place has been reached and in some cases passed, with equipment in existence today. There are probably a few isolated places where still larger crushers could be used to advantage, but he considers them very few in comparison with those where the larger crushers could not be so employed.

[1] Ellis Soper. The Rotary Kiln. The Journal, October 1910, p. 1577.

In Par. 56, Mr. Mason refers to the crushing strains being taken up internally because the rolls act as an anvil. As a matter of fact, the enormous crushing strains do not reach the bearing or frame without being greatly absorbed, since if this were not the case neither would withstand the strains that would come upon it. It is a question, however, whether the gyroscopic action of these enormously heavy, rapidly revolving rolls does not do a great deal more to absorb the crushing strains than the inertia due to the weight of the rolls themselves, irrespective of the rotation.

In Par. 60, Mr. Mason estimates the capacity of the rolls at 55,000 tons per hour, if a solid stream of crushed rock were delivered from the rolls. His method of calculation is practically the same as that used in obtaining the capacity of smaller smooth-faced rolls. In his computation the writer has assumed that these smaller rolls were filled with soft plastic clay which would pass between the rolls with no voids, and he has figured the contents of the ribbon of clay that would be wound through the rolls in a minute or an hour in cubic feet. For rock he has assumed that a certain proportion based on actual practice of this theoretical cubic feet content would be voids, and that the balance would be crushed rock, weighing 100 lb. per cu. ft. Thus for the problem in hand, Mr. Mason states that 18,306 cu. ft. would pass the rolls in one minute. Assuming nine-tenths to be voids and one-tenth rock, as would be the maximum in rolls of this character in normal operation, 1830 cu. ft. of rock would actually pass the rolls, which at 100 lb. per cu. ft., would be $91\frac{1}{2}$ tons per minute, or 5490 tons per hour.

In Par. 61, Mr. Mason describes a means of cutting off the current from the motor driving the rolls to avoid the peak loads. I do not exactly understand how this operates, assuming the motor to be an induction motor, as the current consumed by an induction motor varies inversely in speed, that is, the slower the speed, the greater the current consumed and the power output. Thus, assuming that the motor was not disconnected, the rolls would not be slowed down as much in the crushing action, on account of the power supplied by the motor, as if the motor were disconnected during the crushing action. After the crushing had been accomplished, therefore, in the case of disconnecting the motor when the current was turned on again, the motor and rolls would be rotating at a slower speed than if the motor was not disconnected, and the peak load of current used would be greater until the rolls had been brought up to speed again.

Professor Landis' paper gives the impression that mechanical engineers engaged in the cement industry have not done so much toward developing this industry to the highest possible state of efficiency as they should, or as engineers engaged in other industries have done during the same period of time. Thus the figures given in Par. 26 of 16 lb. of coal required theoretically to burn a barrel of clinker, as opposed to 90 lb. of coal actually used, seem especially bad; but upon considering what these figures mean, and comparing them with the efficiency in the use of coal in railway locomotives, steam power plants, etc., they appear on the contrary to be exceptionally good. For example: a locomotive burning 4 lb. of coal per h.p-hr. at 12,000 B.t.u. per lb., uses 48,000 B.t.u. per h.p-hr., while theoretically it should have used approximately 2540 B.t.u.; or where 90 lb. of coal was burned, theoretically the combustion should have been $4\frac{3}{4}$ lb.

Making the comparison with a gas engine, using 1 lb. of coal per i.h.p. at the same heat value, 12,000 B.t.u. is used, where 2540 should have been used; or where 90 lb. of coal is consumed less than 20 lb. should have been used. When the actual amount of fuel consumed in locomotives and in stationary power plants is compared with that used in the cement industry, the waste by comparison is very much greater in these other industries and the cement engineer as well as the cement chemist, should be congratulated upon making so excellent a showing in comparison with the development of economies in other industries during the same period of time. Greater fuel economies are desirable in the cement industry and I think everyone is working towards this end. Engineers engaged in other industries can unquestionably make themselves valuable, by pointing out methods of economy that have been developed in their own line, with all the details of which those engaged exclusively in the cement industry may not be familiar. The freer the interchange of ideas between engineers engaged in one industry and those engaged in another, the better for both. For example, the question of dust collecting in the cement industry is more or less in its elementary stages, while in the smelting of precious ores it has been carried to a fine point and there are ample data and plenty of examples available.

The point made in Par. 12 of Professor Landis' paper, has already been referred to. Every plant manufacturing cement from rock has a crushing plant in connection with it that is very similar to a great number of commercial crushing plants in existence in this country and the problems are the same in both as regards the size of the largest

crusher. There is no question but that it can be overdone, but this does not seem the tendency in the cement industry. There seem to be very few cement plants that can afford to use a smaller crusher than a No. 18 gyratory with two 36 in. by 8 ft. receiving openings, and unless the rock is mined or quarried by the gloryhole process, or lies in ledges so thin that it is blasted into very small pieces, no cement plant should have a crusher smaller than a No. 12, having two openings, 27 in. wide. While it may be true that only a few pieces of rock will come from the quarry more than 36 in. across, it is equally true that a great many are only a little smaller than this.

In Par. 13 Professor Landis states: "It is only recently that attempts have been made to utilize the hot gases from the kilns to perform the drying of the mix." Professor Landis would seem to be mistaken, as several installations made eight or ten years ago are recalled by the writer, as well as a number made at various intervals since then, and he believes this has been tried a number of times in Germany; while he does not know of any recently made and working with great success. The difficulty is not all in the drying of the rock, as in many cases this can be done easily, but in the hampering of the kiln in its primary function of producing Portland cement. Any attempts to save waste heat that hamper the kiln in this function are ultimately a detriment. This has been the greatest drawback to the utilization of the waste gases. Another thing that must not be overlooked is the fact that these waste gases are absolutely inert and do not contain any burnable gases. The only heat value they possess is that due to their temperature alone, and they put out any auxiliary fire with which they may come in contact since they consist of a very great percentage of CO_2 and N. It would be interesting to know more of the details of the recent installations referred to by Professor Landis in Par. 13, as the writer believes that a way will be discovered eventually to utilize the waste gases, which carry away most of the heat now lost, without hampering the kiln.

Par. 16 deals with the value of the impalpable powder produced by the grinding machines and the loss of this powder occasioned by air separation. Both air separation and fine screening can be carried too far, but either are advantageous up to a certain point, since by taking out a certain portion of this impalpable powder from a grinding machine, it prevents the cushioning of the blow of the grinding parts which results when the machine is largely filled with this fine powder and prevents the grinding of the coarser particles.

Par. 17 relates to the wet process. Unquestionably a great deal

can be learned regarding the efficient use of this process from Germany, since wonderful strides have been made there in the past few years. It is considered at present by some of the eminent engineers to be better than the dry process. While this may be true in Germany, however, this country has made such wonderful strides in the development of the dry process, that it is the full equivalent of and probably a little superior to the modern wet process in Germany.

In Par. 18, Professor Landis refers to 50 or 100 per cent of water in a mass of ground material in the wet process. In Germany at the present time what the thick slurry process is called is running on as little as 25 per cent or less of water.

As regards storage, discussed in Par. 20, the question of shutting down everything but the rotary kilns in a cement mill over Sunday, in order to establish better conditions for the workmen, is just arising. It is becoming more and more difficult to get good operators to work seven days a week all through the season and I believe shuting down the mill on Sundays will ease the labor situation, as well as permit such repairs as are required to be made during the busy season, without loss in time to the kiln.

Par. 22 states that in size these kilns vary from 60 ft. long and 5 ft. in diameter up to 240 ft. long and 12 ft. in diameter. There are kilns in use varying from one size to the other, but the modern cement plant does not install a kiln smaller than 125 ft. by 8 ft., and 150 ft. by 10 ft. would seem to be very much nearer the standard size kiln that will be installed in the next year or so. Kilns smaller than these are now considered obsolete.

In Par. 23 the sizes of kiln stacks are discussed. At least two of the better manufacturers of kilns have now carefully established sizes of kiln stacks and all kilns of any given size have the same size stack, which is properly adapted to the kiln with such minor modifications as might become necessary, depending upon the location of the plant, either in a valley between two hills or upon an open plain.

In Par. 26 reference is made to the saving in heat in the rotary kiln by regeneration in an undercooler. The installation of a rotary cooler immediately under the kiln is an ideal one where it can be made. The writer has, however, been unable to figure out a greater fuel saving than approximately 5 per cent, the amount actually borne out in practice.

Regarding the saving in heat lost by radiation from the kiln shells, without questioning its desirability the writer has found heretofore that the extra investment required to conserve the small amount of

heat that can be saved is too great to warrant going very far in this direction, unless the investors are satisfied with a return approximating 7 per cent or less upon the money invested. A great many people have suggested putting a covering on a rotary kiln *outside* the steel shell. This cannot be done, since in order to prevent the steel from becoming overheated and thereby weakened, the heat communicated to it from the brick lining must be radiated. The only place where insulating material can be put is inside the shell. This applies, of course, to the hot end only and this is the only part of the kiln shells where excessive radiation takes place. As a matter of fact, the question of diminishing the returns upon the investment is really what prevents the installation of a great many of the so-called economies.

The cement business cannot be conducted upon a very narrow margin at the present time on account of the fluctuation of the price. Suppose a cement plant has $1,000,000 invested in it, is earning 20 per cent during good seasons, and the investment of $500,000 more would result in the earning of 7 per cent upon the extra investment. The earning upon the entire investment would then be $15\frac{2}{3}$ per cent, and this extra investment therefore appears undesirable in an industry where the earnings may be decreased 15 per cent by a single cut in the price.

In Par. 32 it is stated that the Lehigh region is producing cement as cheaply as any district in the country and cement manufacturers have to strive hard to cut the total cost at the mill below $0.55 to $0.60 per bbl. In spite of the natural advantages in the Lehigh Valley, the average mill cost is not believed to be so low as in some other districts. There are mill costs running under the figures named, but there seems to be no standard method of calculating the mill costs. Furthermore, the publishing of these mill costs, which are very much lower than the ultimate cost is not fair to the industry, since cement sold at $0.80 per bbl., for example, would seem in consequence to carry a good profit. There are very few plants in this country at the present time that can afford to sell cement continuously for $0.80 per bbl. at the mill. The depreciation of the equipment due to the rapid strides in the industry is an item that seldom shows up correctly upon the cost sheet..

Cement manufacturers are more liberal with the interchange of data among themselves, both as regards costs and details of operation, than any other industry. To that fact alone is due the rapid development and improvement in an industry which has placed the

United States in the front rank of all countries producing Portland cement, while 15 or 20 years ago it was behind all the great Portland cement producing countries of Europe.

ORAL DISCUSSION

FRANK B. GILBRETH asked if the setting of cement could be delayed in any way without injury. It would be very helpful in connection with building construction to be able to add to the mixture at the time it is mixed wet, something that will keep it from setting for four or five hours without injury.

R. K. MEADE[1] replied that several materials can be added for this purpose, of which gypsum is one of the best. Engineers would probably have to use it in the form of plaster of paris. In adding it great care should be exercised and the amount to be employed would have to be determined in each case by experiment. Up to a certain percentage plaster slows the set of cement, but after this it has the effect of quickening it.

Cement clinker as it comes from the kiln will set almost immediately. In the manufacture of Portland cement gypsum is added to delay the setting, but if too much is used the cement will set as quickly as if none had been added. Hydrated lime could also be used, but a larger percentage would be necessary, perhaps even as high as 8 or 10 per cent. Calcium chloride can be used, about the same quantity of this as of plaster being required. Sugar added in small quantities would delay the set also, but this is said to be injurious.

W. R. DUNN, in reply to a question, said that the waste heat from the rotary kilns had been used in several cases to heat water for boilers, but that it was more frequently employed to dry raw material.

Mr. Dunn stated that the loss of cement dust from the stacks of the rotary kilns could be largely but not entirely prevented by the use of large dust catchers at the end of the kilns. His practice showed this loss to be in the neighborhood of 2 per cent and at times as low as 1 per cent.

H. STRUCKMANN said that experiments had shown that from 5 to 7 per cent of the material was lost through the stacks, and that it had become necessary to prevent this loss, especially where the cement

[1]General Manager, Tidewater Portland Cement Co., Baltimore, Md.

plants are located in thickly populated districts. A solution of the difficulty was to be found in the adoption of the wet process. He had never seen a plant employing the wet process discharge as much dust from its stacks as the average dry plant.

The problem of making a plant operate under the wet system as efficiently as the dry plants is comparatively simple, and the dust nuisance is here practically eliminated.

The small amount of dust lost in the wet process can be reduced to a minimum by employing proper methods for injecting the slurry into the kiln. At the present time the slurry is delivered to the kiln in a solid stream, permitting the escaping gases to carry part of the raw material, from which the water has been driven off, with it. If spray nozzles are used, similar to the method used for injecting water into jet condensers, the gases on their way to the stack would pass this spray of slurry, which would arrest the dust and very materially lower the temperature of the waste gases, consequently increasing the efficiency of the kiln.

R. K. MEADE agreed that a spray nozzle would increase the efficiency greatly, but he thought it would increase the loss of dust. He believed that more careful attention to the design of kilns and stacks would help to eliminate dust with the dry process. The opening from the kiln to the stack is often made too small, and the velocity of the gas is thereby greatly increased just at the point where the raw material is admitted. With lower velocity of gas at this point and larger dust chambers at the base of the stacks, much less would be lost than is now the case. The discharge of the raw material near the bottom of the kiln instead of at the middle should tend to reduce this loss also.

C. J. REILLY[1] stated that he had been connected with both the wet and the dry processes for a number of years, and agreed that the material does not become dry within 15 ft. of the point where it is admitted to the kiln. In the case of one of his plants, located next to a factory where newly varnished furniture was placed close to the windows, it became necessary to eliminate the dust nuisance entirely. After trying many other methods, the problem was solved by putting an ordinary cactus spray into the stack and using a small quantity of water under ordinary pressure. As the gases pass through the

[1]Supt., Sandusky Portland Cement Co., Syracuse, Ind.

spray, the dust is caught and taken to the bottom of the stack as sediment. This method has entirely eliminated the emission of dust from the stacks and the consequent loss of finely ground raw material.

G. P. HEMSTREET thought that the inference might be drawn from the paper by Mr. Griffiths that the shaft and wheel drive was to be preferred for a cement plant, or for any similar process which might be described as a chain process, that is, where the material starts at one end and goes through a series of machines, a stoppage of any one of which would hinder the entire process.

About two years ago he was about to design a large stone-crushing plant, and had under consideration both the shaft-driven and the electrically-driven plant. In his visits to a number of cement and stone-crushing plants and installations of coal-handling machinery, he found quite a difference of opinion regarding the relative merits of the two types. The majority of the men in charge of these plants, however, favored the electrically-driven plant, while Mr. Griffiths seemed to believe that this type is not as efficient on the B.t.u. basis as the belt or rope-driven plant.

In a chain process there are other things to consider besides the actual operating efficiency. For example, on a large lineshaft several feet long, containing a number of clutches, pulleys and rope drives, any trouble with one of the drives will necessitate the stopping of the entire shaft or the throwing out of a large clutch. The speaker mentioned the many difficulties experienced with clutches, and and asked the experience of others present regarding the value of such installations as compared with electrically-driven plants.

F. L. SCHWENCK[1] stated that the manufacture of Portland cement from blast-furnace slag had become quite prevalent in Europe, and asked whether American engineers regarded this as equal to Portland cement made from other materials.

WM. M. KINNEY[2] replied that two kinds of cement were being manufactured from granulated blast-furnace slag. One, Puzzolan, or so-called slag cement, which is a mixture of granulated blast-furnace slag and slaked lime; the other, a true Portland cement, in the manufacture of which a definite proportion of granulated blast-furnace slag and limestone is first finely pulverized, then burned at a high

[1]Burrakur Iron Wks., Kulti, E. I. Rwy., Bengal, India.
[2]Asst. Inspc. Engr., Universal Portland Cement Co., Pittsburgh, Pa.

temperature (close to 3000 deg.) in rotary kilns. The resultant hard clinker is ground to powder. The process of manufacturing Portland cement from slag and limestone is identical with that manufactured from natural deposits.

Over 7,000,000 bbl. of Portland cement manufactured from blast-furnace slag and limestone were used in the United States last year, being more than 9 per cent of the entire Portland cement output of this country. This material in every way fulfils the standard specifications for Portland cement of the American Society for Testing Materials and those given in professional paper No. 28 of the United States Army Engineers.

W. S. Landis asked whether the power required for grinding wet, plus that required to pump the mixture, plus that required to stir it up in the agitator, is equal to the power required for dry grinding.

H. Struckmann in reply said that a number of tests had shown an average saving of from 30 to 35 per cent in power by grinding wet, besides a much better fineness of the material.

A great many plants in the West use shale, wet clay or chalky limestone for raw material, containing from 15 to 20 per cent of moisture in the natural condition. To drive off this moisture preparatory to grinding the material dry requires as high as 300,000 B.t.u. per bbl. In such cases there is no question but the wet process is the more economical, provided the kilns are long enough to take care of the burning as well as the drying process in one operation.

The relative merits of the wet and dry process may be illustrated by the development which has recently taken place in Europe, when practice has been directed towards fuel economy, the most important item in the cost, and where the gradual gain of the wet process would indicate its merits.

C. J. Reilly referred to the statement in the paper by Professor Landis regarding the abandonment of the air separator in grinding raw materials, agreeing that the old method of separating by air is obsolete and worthless for the purpose there intended, but citing a case in which he had found a new method quite useful. In grinding a hard Trenton rock and silicious clay so that 92 to 94 per cent passed a 200-mesh sieve, it was found impossible in the ordinary way by using screens to get the output with a reasonable amount of power. An experimental plant was installed, and air separation tried in con-

nection with a standard mill, with the result that the grinding was found to be as good as on any modern mill, although the power consumption was 35 per cent less than has yet been accomplished in the cement industry. The new method has improved the quality of the material over that coming from the tube mills, this new device being applied to the Fuller mills and entirely eliminating the tube mills formerly used for the purpose.

W. B. Ruggles said that he had recently designed a 3000-bbl. mill in which all the materials were separated by air. Ten Raymond mills were installed and gave a product of 96 per cent through a 100-mesh sieve and 91 per cent through a 200-mesh sieve. The cost of quarrying, transporting the materials through the mill on an aerial tramway, crushing, storing, removing from storage, drying, grinding with Williams mills, pulverizing with Raymond mills, and delivering to the bins of the rotary kilns is $0.71 cents per ton.

F. H. Lewis. At the end of any given period in the operation of a cement plant there are two results of operation which can be accurately determined; (*a*) the barrels of cement produced; and (*b*) the number of tons of coal used at the steam plant. There is one detail of operation which is extremely difficult to determine with any degree of accuracy; that is, the total number of h.p-hr. developed by the steam plant. The writer has never seen this accurately recorded in a direct-driven steam plant. It can be done with recording wattmeters and a sufficient amount of labor in an electric plant; but if there are any reliable data of this kind for cement plants, they have not come to the writer's notice.

Taking Mr. Griffiths' figures for coal used per bbl. of cement as given in Table 4, we find steam coal per bbl. of cement for: plant No. 1, 107 lb.; plant No. 2, 69 lb.; plant No. 3, 71 lb.; plant No. 4, 200 lb.; and plant No. 5, 93 lb. This gives us data based on results of operation which are regularly accounted for in cement bookkeeping; and also results which can be compared with other similar data. The figures for plants Nos. 2 and 3 are average normal results; for plant No. 1 the figures are high, but also normal for a plant of the character described by Mr. Griffiths. For plant No. 5 the figures are high, while for plant No. 4 the cost of boiler fuel is ruinous. It is quite certain that the best steam plants are producing a barrel of cement with less than 60 lb. of boiler fuel, and electric plants with less than 70 lb.

When we consider Mr. Griffiths' Table 3, however, we find the coal per h.p-hr. to be high, even for plants Nos. 2 and 3 which are operating successfully at full output; and the h.p. per bbl. of cement for these plants to be low. That is, the product of these two factors is normal, but the subdivision appears to have been made on an assumption which gives high figures for coal and low figures for power. It is no uncommon thing in cement plants for engines to carry 10 per cent to 40 per cent over their rating. As a general proposition cement-plant engines operate with loads varying frequently within short intervals, from no load, or friction load, to an overload. The difficulty of arriving at the h.p-hr. under such conditions is apparent and probably has not been sufficiently considered by Mr. Griffiths.

CLOSURES[1]

W. S. Landis. The cement industry, like most others of similar widespread activity, does not permit of standardization, that is, a standard mill could not be designed to fit all conditions of raw material supply, and it is thus that the problem of the wet and the dry processes has arisen. In the Lehigh region with the supply of an almost homogeneous raw material, the dry process is used exclusively. In certain other districts of the country the raw material is of such irregular composition that some sort of receiver for adjustment of the mix seems almost a necessity and the wet or semi-wet process with its slurry tanks offers a solution.

The precious metal industry has furnished us with abundant information on the advantages of wet crushing and grinding, and Mr. Struckman has given in his discussion of my paper data of direct application in confirmation of these facts, regarding fuel saving and fine grinding, which I consider to be the most valuable contribution to the industry in many years.

Admitting now that a semi-wet process can operate with approximately 35 per cent water and that the raw materials as received at the mill contain 5 to 10 per cent water, the question arises, is it most advantageous to dry this material and grind it, or to add 20 to 30 per cent more water to it and then dry it in the kiln; or is the total cost of agitating, pumping and drying in the kiln less than that of advancing to the same stage of product in the dry mill, allowing, of course, for the difference in grinding power? An engineer recognizes as absurd the statement often made that the evaporation of the water in the kiln takes no fuel because the gases leaving the

kiln are hot anyway. Any heat used in the kiln for drying could be employed just as advantageously in raising the temperature of the dry material as in evaporating water. I am sure that all would advocate the wet process did the raw material as it is taken from the ground contain the 30 to 35 per cent water demanded by the process, but where water is actually and intentionally added, a different phase is presented. And, to complicate the question still further, we must note that certain successful mills are receiving wet raw material, drying it completely, grinding, adding water and pumping to slurry tanks for further treatment, as in a true wet process.

I think Mr. Van Zandt's comparison of fuel economy in the cement industry and in standard power practice is somewhat overdrawn. A good steam boiler will put 75 per cent of the heat value of the coal into steam. The steam engine on the other hand, transforms into work only about 20 per cent of the heat in the steam supplied, the difference, the latent heat of the condensation of the steam not being transformable into work in the steam engine. Thermodynamics teaches us that this latent heat of condensation, amounting to approximately 60 per cent of the calorific power of the coal burned under the boiler, will never be available to the steam engine, no matter to what stages engine design may be developed. If Mr. Van Zandt will leave out of the question this latent heat and examine the work of the mechanical engineer with respect to the transformation of the rest of the heat he will have a different comparison. The latent heat (reaction heat) in the case of the cement kiln amounts to only 18 per cent of the calorific value of the coal burned.

In the case of the gas engine, the comparison is more favorable. A good gas producer will put 85 to 92 per cent of the heat value of the coal burned into gas. The engineer has been too busy, however, perfecting an engine that will run to spend much time on the economical side. I firmly believe that in the next ten years the efficiency of the gas engine and accessories will be very markedly raised.

Mr. Van Zandt in discussing the use of the waste kiln gases for drying the mix has called attention to the failure of the system advocated in the paper. As this question has been brought up so often and the failures have been so pronounced it demands attention. In the installations made years ago an ordinary dryer was connected in series with a kiln. The draft of the kiln was so hindered that fuel could not be burned at the desired rate and the output was

thereby greatly reduced. What else could be expected where a 4-ft. dryer is connected to the end of an 8-ft. kiln (figures somewhat exaggerated), the drier being heavily loaded with rock dropping through the free space leaving no room for the great volume of kiln gases? The free space in the drier should be even larger than that in the kiln if the draft is to be preserved. Had a 10-ft drier of suitable length been used instead of the smaller one no trouble would have been experienced. There is such an installation of drier and kiln successfully working in the Lehigh region.

Mr. Van Zandt's reference to the fuel saving by the use of an undercooler opens up an interesting question. The simple undercooler costs, in many cases, less than the elaborate cooling towers so often erected to dispel the heat in the clinker to the surrounding atmosphere. Thus its use, even if it saves only 5 per cent (I have figured it out higher than this) of the fuel is worth consideration. Again, if the cost of the plant is a serious consideration, there is the alternative of doing away with either system and discharging the clinker directly from the kilns into a car fitted with a water spray. This will quickly dissipate the heat, slake the "free" lime and so avoid the necessity of aging, and consequently soften the clinker, saving an enormous amount of power in the finishing grinding. If properly handled the clinker can be made to dry itself. I have never quite understood why more general use of this scheme was not made.

W. H. Mason. I certainly do not agree with Mr. Van Zandt in his belief that the largest rock which can be economically handled is one that can pass through a steam shovel dipper. In my opinion, the largest rock that can be economically handled is of a size such that it can be manipulated by the crusher. The danger of damaging cars from the teeth of the steam shovel is not serious, if the car is properly designed for this service. The writer's experience has been that 20 cars are damaged by ordinary train movements to one that is damaged by loading large stone on the cars.

Referring to the tachometer test, I would state that no two of these show the same result, as would be expected. It is usual for the slugger roll to be reduced somewhat more in speed than the regular roll; but this is not always the case, for occasionally the regular roll delivers a slugging blow and shows a proportional reduction in speed. It is also common to have large stone so shattered in one or two slugging blows that all the remaining stone will come within the

angle grip of the rolls, and apparently this is what took place on chart *K*, shown in Fig. 7.

Referring to Mr. Van Zandt's suggestion that the peak load would be greater if the circuit breaker threw off the current during the crushing period, I would state that the recording watt-meter connected to the motors driving these rolls does not bear him out.

L. L. GRIFFITHS. Mr. Lewis loses sight of the fact that the following conditions affect fuel or coal consumption irrespective of whether the power plant is used in connection with a cement plant or some other industry. First we have the B.t.u. value of the fuel, second the chemical composition of the fuel especially with reference to the ash forming elements, third, the type and design of the furnace in which the fuel is used, fourth, the efficiency of the prime mover, fifth, the mechanical efficiency of the arrangement of the power transmission machinery, and sixth, the physical condition of the machinery operated.

The designs of different general classes of grinding machinery in the cement mill also vary widely with reference to the barrel output compared with the power consumption, all of which must be considered.

Mr. Lewis' further criticism that the figures for coal used were those accounted for by the bookkeeping department of the cement plant is correct; but in this connection I may say that they are the ones on which money settlement is made and that the money investment is what affects the cost per barrel.

In the direct-connected plants cited in the paper the power output of the engines was determined by indicator cards taken daily on each shift. The possibility of error is present, but the fact that the figures used in compiling the data given in the paper were from yearly results and not short test periods makes the possible error negligible.

With reference to the electrically-driven plants the power plant output is accurate as both these plants are equipped with continuous recording watt-meters.

Mr. Lewis is not correct in his statement that the loads in a cement plant vary within short intervals, as it is generally recognized that the cement plant gives a very uniform power curve, varying only between the day and night shifts by the difference in power requirements of the store house, which usually operates but ten hours per day, and the lighting load, which is thrown on at sunset.

If all the conditions set forth in the paper are considered, the reason of the varying fuel consumption per barrel in the group of plants given will be readily seen.

SYMPOSIUM ON INTERCHANGEABLE PARTS

No. 1312 *a*

THE ASSEMBLY OF SMALL INTERCHANGEABLE PARTS

By John Calder, Ilion, N. Y.

Member of the Society

In the branches of mechanical engineering and manufacturing devoted to complicated and highly finished business and domestic appliances, as distinguished from the many simpler and rougher forms of hardware, the direct labor costs of assembling the numerous small interchangeable parts are very important items. They often considerably exceed the whole cost of the material and the expense incurred upon them is much more within the control of the management.

2 Long before any proposals were made towards the general modification of the day-rate system of labor reward in most plants, the large industries mentioned had been driven by necessity to cheapen and intensify production by methods more or less systematic.

3 The movement in mechanical engineering in recent years towards scientific time study of labor tasks owes much to the proceedings of the Society, particularly to the contributions and well-considered generalizations of Messrs. Fred. W. Taylor and H. L. Gantt. It is now fully realized that the scientific principles advocated are simply common sense organized to a high degree and that what is needed is not so much expert advice and staff and equipment changes as a firm realization of the possibilities by an efficient management and the ability successfully to inoculate the existing organization with like optimism.

4 This movement found many interchangeable parts industries in a position very favorable for profiting by it and not a few of them have resumed the economical evolution of their processes and labor methods

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

along the new lines at a rapid pace after a period of more or less stagnation. They have accomplished this in many cases by adopting the new principles in full, but using the existing staffs and appliances and dispensing with the capital outlays and outside expert aid which is sometimes absolutely necessary in machine shops when the change of system is very marked.

5 Table 1 is a typical organization chart for a works employing several thousand people in the interchangeable parts industry when the great bulk of the parts production is assembled into finely adjusted machines. It is the purpose of this paper to outline and illustrate briefly the principles upon which the parts assembly department is conducted, but they are equally applicable to all the manufacturing branches indicated on the chart.

6 Economical parts assembly implies, among other things, the securing for such operations in shop practice at the initial stages in the history of a new part of

a The highest assembly speed, and

b The prescribed quality of work; neither more nor less.

7 If these conditions are fulfilled, the best economy will be most quickly attained when

c The classes of labor available are properly selected, graded and trained.

8 Economical assembly is sometimes attained by indirection, after a long drawn-out and intermittent process of elimination, trials and modifications based upon rough occasional observations by foremen and others of ordinary workers in the shop. This course is costly, uncertain and slow, and early expensive and inefficient conditions are apt to be perpetuated.

9 Starting with small interchangeable parts of the most practicable form, the management of such a works is in a position to determine for itself, without the guesswork sometimes necessary in complicated machine tool operations, the best assembly arrangements of apparatus and of labor. Any plant doing light repetition work in quantities can afford to devote a small inexpensive department entirely to this task and to cease the guessing methods, whether aided by premiums or not, which often do duty for painstaking time study and efficient initial criticism of apparatus and routine.

10 Certain preliminary precautions and results in the machine shop and other departments producing the unit pieces to be assembled are necessary, such as

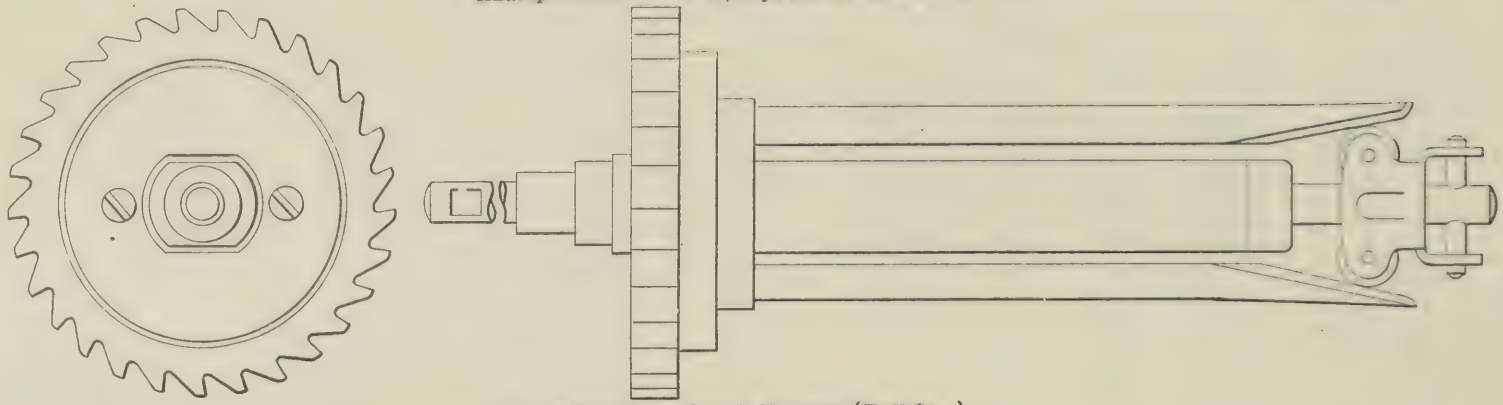
d The designing of the form of the part with quick assembling in view.

TABLE 1 ORGANIZATION OF LARGE WORKS MAKING AND ASSEMBLING SMALL INTERCHANGEABLE PARTS

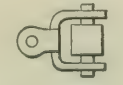
- MANAGER
  - EXECUTIVE DEPARTMENTS
    - GENERAL OFFICE
      - Correspondence
      - Cashier and Bookkeeper
      - Industrial Betterments
        - Works "Bulletin"
        - Suggestion System
        - Foremen's Club House
        - Works Band
        - First Aid
      - Time Records and Payroll
      - Rate Records
      - Costing and Inventories
    - PURCHASING
      - Purchasing Raw Material and Supplies
      - Receiving Clerk, Teams, R R. Siding
      - Raw Material and Supply Stores
    - FINISHED PARTS AND PRODUCTION ORDERS
      - Finished Parts Stock Room
      - Issues all Production Orders for Parts
      - Issues all Parts for Machine Manufacture
    - SHIPPING
      - Stock of Finished Machines, Packing Room
      - Shipping and Billing Machine and Part Orders
    - WORKS ENGINEERING
      - Light, Heat, Power, Ventilation
      - Millwrights, Electricians, Plumbers
      - Fire Corps, Watchmen, Yard Labor
  - MANUFACTURING DEPARTMENTS Superintendent
    - MANUFACTURE OF PARTS Asst. Supt. and Foremen
      - Pattern Shop
      - Brass Foundry
      - Iron Foundry
      - Japanning
      - Tinsmiths
      - Drop Forging
      - Punch and Press Work
      - Automatic and Hand Screw
      - Annealing and Hardening
      - General Grinding
      - Light Milling and Drilling
      - Cast Iron Milling and Drilling
      - Polishing
      - Plating and Buffing
      - Wood and Rubber Working
      - Assembling Compound Parts
      - Tool Designing and Model Making
      - Tool Making
      - Inspecting Parts
      - Testing Rates and Methods
    - MANUFACTURE OF MACHINES Asst. Supt. and Foremen
      - Assembling Machine Group No. 1
      - Assembling Machine Group No. 2
      - Assembling Machine Group No. 3
      - Assembling, Final Stage
      - Inspecting Machines
      - Adding Special Fixtures
      - Rebuilding Machines
      - School of Repairing

TABLE 2 DIVISION OF LABOR AND SPEED OF WORKERS IN ASSEMBLING SMALL INTERCHANGEABLE PARTS

Examples where women, boys and men are used



Assembled Piece Clutch No.31531 (Full Size)

Stage No.	Unit Parts 18 in Number Assembled in 10 Stages	Operations in Sequence	Grade of Labor	Average Time Including Stops, from Records of 100,000 Pieces	
				Min.	Sec.
1		Ream 2 holes 0.073″ in Roll Holder No.30670 on speed lathe	Women	0	7
2		Assemble roll holder No.30670 2 rolls No.30680 and 2 pins No.67640 at bench	Boys	0	45
3		Ream 1 hole 0.166″ in clutch frame No.30361 on speed lathe	Women	0	5

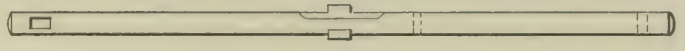
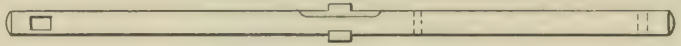
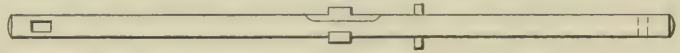
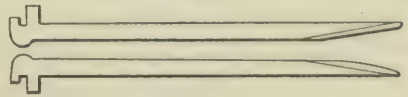
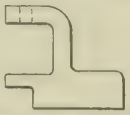
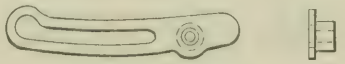
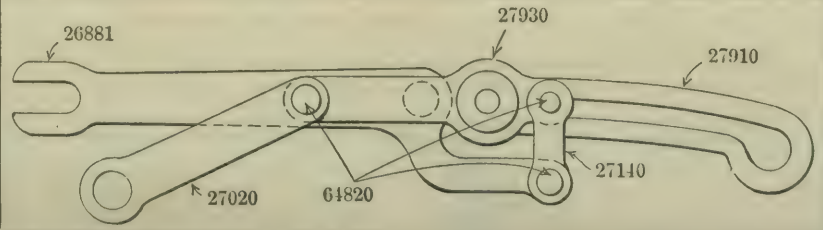
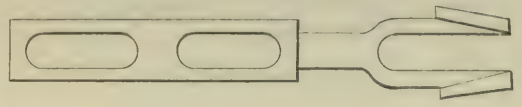
4		Tap 2 holes 0.08″–56 in clutch frame No.30361 on tapping machine	Men	0	16
5		Ream 2 holes 0.085″ and 0.059″ in shaft No.31620 on speed lathe	Women	0	10
6		Straighten shaft No.31620 on bench block	Boys	0	8
7		Put pin No.65610 in shaft No.31620 with bench fixture	Boys	0	20
8		Put in washer No.90000 and assemble tube No.31521 and spring No.67210 at bench	Men	2	24
9		Bend and gage clutch jaws No.30690 in bench vise	Men	0	31
10	See full-size sketch	Assemble clutch jaw No.30690 ratchet No.30370, frame cover No.30330, 2 screws No.70402, and roll holder comp., with pins No.64650 at bench	Men	3	52
	The direct labor cost of assembling 1 clutch No.31531 is 3.784 cents	Average time to assemble 18 unit parts to form clutch No.31531	Women Boys and Men	8	40

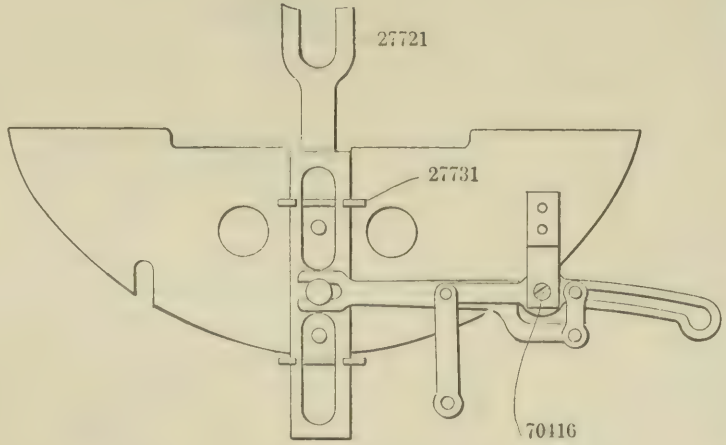
TABLE 3 DIVISION OF LABOR AND SPEED OF WORKERS IN ASSEMBLING SMALL INTERCHANGEABLE PARTS

Examples where men only are used

Unit Parts 15 in Number Assembled in 9 Stages	Stage No.	Operations in Sequence	Grade of Labor	Average Time Including Stops from Records of 100,000 Pieces	
				Min.	Sec.
	1	Ream 1 hole to size 0.101″ in support bracket for ribbon driving arm No.27060 on speed lathe	Men	0	10
	2	Ream 1 hole to size 0.101″ in ribbon position adjusting arm No.27910 on speed lathe	Men	0	10
	3	Ream 1 hole to size 0.101″ in ribbon shifting arm No.27930 on speed lathe	Men	0	10
	4	Burnish adjusting arm No.27910 on bench	Men	0	12
	5	Assemble driving arm, No.26881 shifting link No.27020, connecting link No.27140, stud for adjusting arm No.27150, adjusting arm No.27910 shifting arm No.27930 and 3 rivets No.64820 on bench	Men	0	48



6	Bend and gage carrier No.27721 to bench fixture	Men	1	12
7	Assemble support bracket No.27060 and guide No.27731 with 2 screws No.74410 on bench	Men	1	2
8	Fit carrier No.27721 and guide No.27731 on bench	Men	1	36
9	Final assembling of riveted links with carrier No,27721 and guide No.27731 with screw No.70416, on bench	Men	1	58
The direct labor cost of assembling No.27710 is 4.195 cents	Average time to assemble 15 unit parts into 1 carrier guide and driving arm No.27716	Men	8	18



Assembled Piece No.27710

(Half Size)

TABLE 4 DIVISION OF LABOR AND SPEED OF WORKERS IN ASSEMBLING SMALL INTERCHANGEABLE PARTS

Examples where women only are used

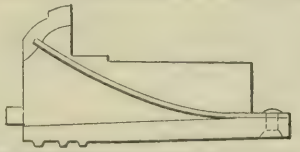
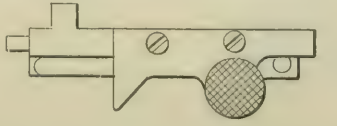
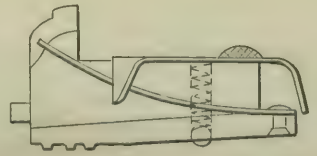
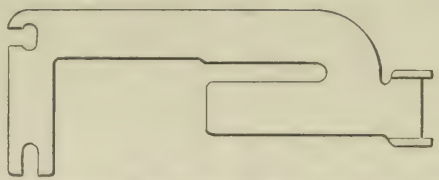
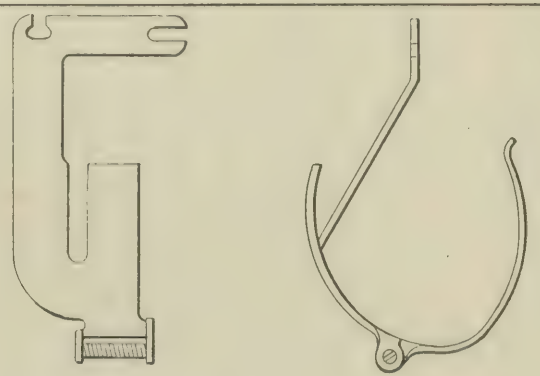
Unit Parts Ten in Number Assembled in Three Stages (Full Size)	Stage No.	Operations in Sequence	Grade of Labor	Average Time Including Stops from Records of 100,000 pieces
	1	Assemble stop No.36202 and spring No.66220 with rivets No.64950 on Bradley hammer.	Women	Min. 0 Sec. 45
	2	Drive pin No.64750 in stop No.37612 on bench.	Women	0 12
	3	Assemble locating ball No.16100, spring No.67150, indicator No.36180 with screws No.70000 on bench.	Women	1 36
	The direct labor cost of assembling one margin stop No.37612 is 4 mills	Average time to assemble 10 unit parts to form one margin stop No.37612	Women	2 33

TABLE 5 DIVISION OF LABOR AND SPEED OF WORKERS IN ASSEMBLING SMALL INTERCHANGEABLE PARTS

Examples where boys only are used

Unit Parts Four in Number Assembled in Two Stages	Stage No.	Operations in Sequence	Grade of Labor	Average Time Including Stops from Records of 100,000 pieces
	1	Tap one hole 0.100″-48 in support No.26571 on tapping machine.	Boys	Min. 0 Sec. 6
 (Half Size)	2	Assemble support No.26571, tension finger No.26580, spring No.67400 with screw No.70838 and gage by weight test on bench.	Boys	1 15
		The direct labor cost of assembling one No.27520 is 6½ mills. Average time to assemble 4 unit parts to form one spool guard No.27520.	Boys	1 21

e The designing and proper maintenance of efficient tools, jigs and fixtures, working to practicable limits in the dimensions and fits.

f An efficient parts inspection service, securing in every department adherence to these limits and the rejection of all defective unit parts.

g The adoption for each unit piece of a shop routing in the machine and other departments which will produce the fitting surfaces in the conditions permitting the least work in assembling. The total abolition of the file or other cutting tools at the bench should be aimed at.

11 If the foregoing conditions are observed, the speed of assembly will then depend upon

h The extent to which the division of labor is carried.

i The thoroughness of the time study of skilled demonstrators in each division, including the elimination of lost motion and the criticism of the machined unit pieces.

j The efficiency of the mechanical facilities provided by the management for aiding the work of assembly.

12 If the above ten conditions (*a* to *j* inclusive) are by suitable organization fulfilled for any new piece before it is presented to the workers in any way and the assembly work done is accepted only after passing an efficient inspection, a straight piecework system of remuneration will often be found best in the interests of employer and employee alike.

13 After such a close preliminary study and exhaustive development of the possibilities, there is no such margin remaining as obtains in the rough approximations of premium setting where the processes are often left to evolve slowly and uncertainly at the initiative of the workers and overseers alone. The employees, therefore, may well be given the whole of their gains on the prescribed standard task.

14 Tables 2 to 5 show a few examples of the division of labor and speed of workers on straight piecework tasks in the assembly of small interchangeable parts, where all the conditions essential to the highest economy have been thoroughly and expeditiously worked out before the assembly task is presented to the manufacturing department. Table 2 gives the final arrangements as determined by the task and method department for a mechanical assembly using the labor of men, women and boys, while Tables 3, 4 and 5 are examples of three mechanical tasks performed by men, women and boys separately.

15 Owing to the minute division of labor and the elimination by careful study of all lost motion, most of the tabulated assembly stages, it will be noted, are performed in much less time than it takes to describe them. About three-quarters of a million unit pieces of several thousand varieties are handled daily on the principles laid down, passing successively through most of the 27 manufacturing departments indicated in Table 1. The total operations involve the use of about 20,000 straight piece rates, each based upon the results of exact preliminary time and method studies.

16 *Apparatus.* The course followed in attaining such results varies according to the nature of each department, but the main features are the same for all and will be sufficiently indicated, the

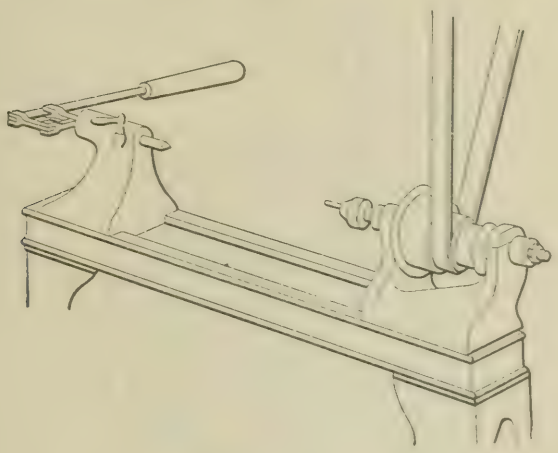


FIG. 1 POWER SPEED LATHE

writer believes, by describing the concrete parts assembly tasks of Tables 2 to 5. In addition to an ordinary bench vise, a 4-oz. hammer, a bench anvil block and screw-drivers, the rest of the tools used consist of simple machines and fixtures, and embrace the following: a power speed lathe (Fig. 1), a vertical power tapping machine (Fig. 2), a bench pin-insertion fixture (Fig. 3), a Bradley hammer (Fig. 4), a horizontal power tapping machine (Fig. 5), a bench gage fixture (Fig. 6).

17 It will be noted that the file is absent from the above list. The possession or use by any assembler of this mechanical persuader toward a fit is forbidden, unless after due cause shown and under special permit. The writer has found that such a prohibition will often throw a considerable amount of new light upon the real condi-

tion and suitability of the machined surfaces prepared in the other shops and which have passed the prescribed gage tests. The general tendency of this course is to raise the whole level of machine shop practice and to help greatly in locating more definitely for the tool designer and maker needed refinements and corrections in the jigs and fixtures, as well as in other processes.

18 *Method.* Prior to the introduction into the shops of any new assembly operations, such as those described, the prescribed apparatus, which is the best that the collective experience of superintendent, foreman and tool engineer can devise, is handed over to the time and motion study department, the few expert employees of which proceed to operate with it on a manufacturing scale. They handle large

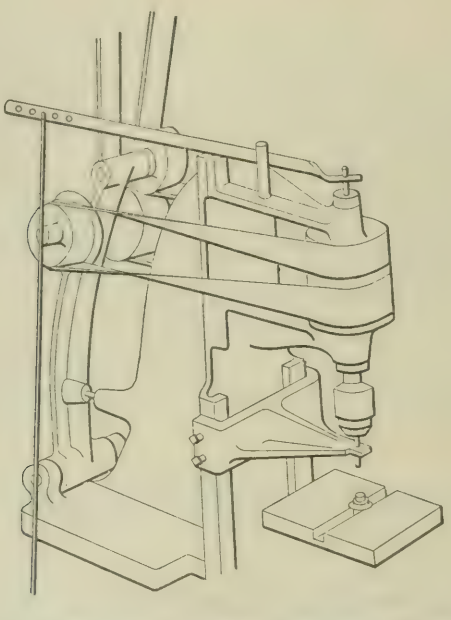


FIG. 2 VERTICAL POWER TAPPING MACHINE

quantities of product which is subject to all the usual inspection tests and passes into the manufacture. In this way the cost of the time study department is reduced to a very small figure, for the working time of its experts is fully as productive as that of the shop. These few experts have been selected and organized from the ranks of ordinary employees in the works who have shown considerable versatility and the power of comparative observation. When such initial criticism of the apparatus and parts supplied as has been well taken has been put into effect, it is the duty of this department to ask and answer quickly by expert demonstrations the following questions regarding

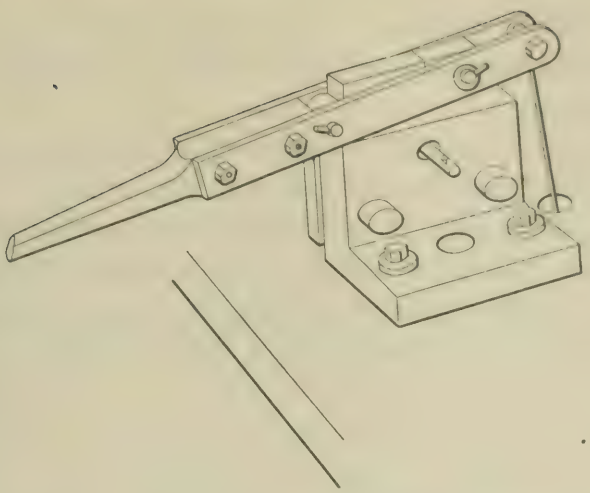


FIG. 3 BENCH PIN-INSERTION FIXTURE

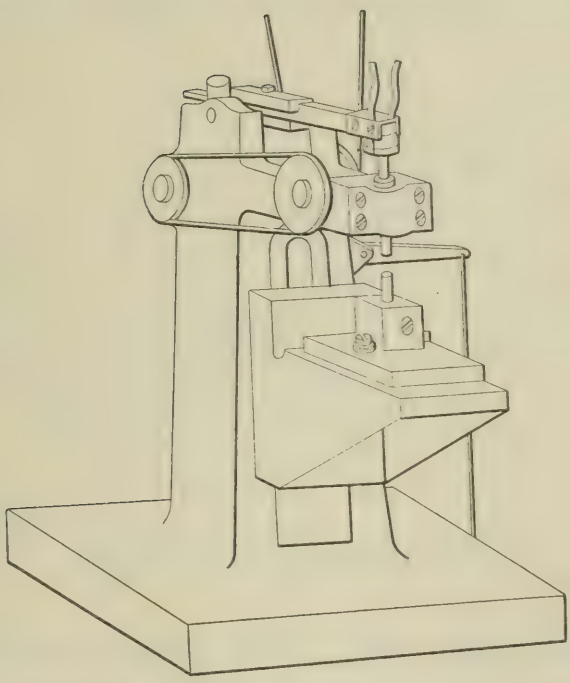


FIG. 4 BRADLEY HAMMER

the particular operations cited, the motions being carefully studied as to their necessity and timed as to their duration:

a Best height and position of the seat or chair for the operator (male or female) at the bench or machine.

b Best position in front, or on right or left of operator, of the several stocks of each unit piece comprised in the assembly task.

c Best position of the hand tools and appliances which the operator has to pick up and apply and lay down again.

d The advantage, if any, which an ambidextrous operator has at any stage, and the cultivation of that faculty.

e Best arrangement for bringing forward and removing the work without interrupting the assembly process.

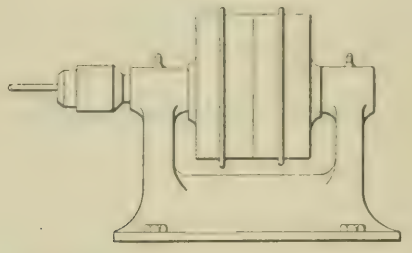


FIG. 5 HORIZONTAL POWER TAPPING MACHINE

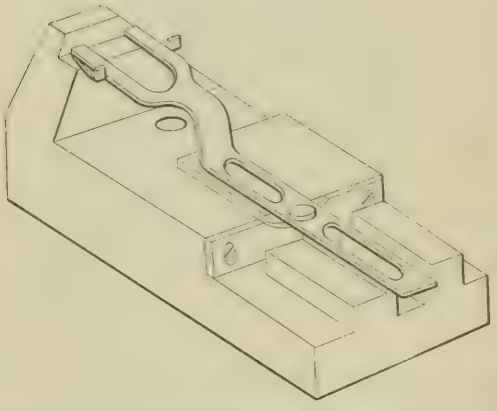


FIG. 6 BENCH GAGE FIXTURE

f Progressive time study under the best conditions of each of the movements finally decided upon as necessary and the combination of all of them on the principle of "least work."

When the possibilities are fully in view a straight piece-rate is fixed, based on these, which will give an industrious worker of average ability in each grade the normal earnings, while the exceptional operator receives the full amount of his surplus.

19 The system described in this paper, for a few tasks of one department only, is applied to all the manufacturing divisions, with details appropriate to their needs. It is also applicable in kind, if not in degree, to economizing many functions in the executive division. There is but one study staff of six members for the whole works. These soon record standardized conditions for every class of manual and machine operation and their work on any new problem or revision is quickly and economically performed.

20 The writer does not favor the restriction of the ordinary shop foremen solely to productive supervision but prefers to encourage them to compete freely with the time study staff for the best result. If pains are taken to give publicity to all good suggestions adopted from foremen a surprising amount of very profitable coöperation can be obtained from them, which enhances instead of diminishes their administrative ability.

No. 1312*b*

THE PROCESS OF ASSEMBLING A SMALL AND INTRICATE MACHINE

By Halcolm Ellis,[1] Newark, N. J.

Non-Member

21 The machine here described is little larger than an ordinary typewriter, but the number of parts is greater and the mechanisms included within it are much more complicated. It is known as the Ellis Adding-Typewriter, and is composed of about 3400 pieces

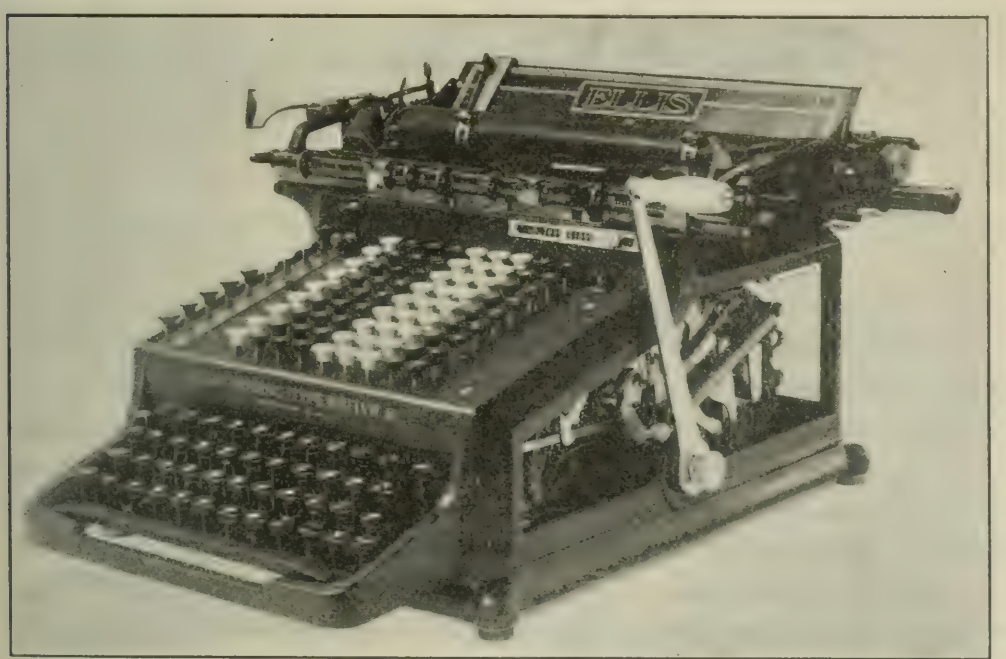


Fig. 7 The Ellis Adding-Typewriter with Glass Sides Removed

assembled in various ways. Before taking up the question of the assembling, it may be well to give a brief description of the machine itself and what it will do.

[1] General Manager, Ellis Adding-Typewriter Co., Newark, N. J.

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

22 The machine (Fig. 7) is primarily a typewriter, and includes all of the elements which go to make up any other standard form of visible printing typewriter. It is provided with a universal keyboard having 42 keys, shift-keys for capitals and signals, tabulating devices and keys for operating them, automatic ribbon movement and ribbon reverse, a device for using a bi-chrome ribbon, thereby enabling the operator to do the printing in two different colors, and a large number of little mechanical conveniences for helping the operator in a speedy execution of his work.

23 Also incorporated in the machine is a complete adding and listing machine, provided with a keyboard containing 81 numeral keys for the setting up of numbers to be added and nine operation keys for setting the machine to perform the different operations of calculation for which it is designed. The adding mechanism controls its own printing device for the printing of numbers, and all totals and sub-totals or other results of arithmetical computations are printed automatically on this machine. The typewriting type and the adding type print on the same horizontal line, and the printing from both is entirely visible.

DESIGNING THE MACHINE TO SECURE RAPID ASSEMBLY

24 The need of a very definite system in the designing of a machine of this character, in order that it may be assembled at all, is evident, and it is also clear that it would be a commercial impossibility to assemble it as a whole, i.e., if each machine were to be built up, piece by piece, by the same man. The solution of the assembling problem lay in the subdivision of assembling and this was accomplished by dividing the machine into sections. These sections in turn were divided into sub-sections, small groups of assembled parts, assembled parts and individual pieces. The machine as a whole was divided into 16 sections, and each section was given a letter; then to keep the symbol numbers in order the individual parts were numbered according to the following schedule:

SECTIONS OF ELLIS ADDING-TYPEWRITER

A = Accumulator Mechanism
B = Base, Frames, Case, Spring Barrel, Carriage Ways
C = Carriage
D = Left-hand Operating Parts, Governor
E = Escapement

F = Tabulating Mechanism and Bell, and Line Lock
H = Holder for Paper Roll
I = Ink Mechanism
K = Keyboard and Parts for Adding Mechanism
L = Left-hand Controlling Parts
O = Operating Handle, Shafts and Parts connected therewith
P = Printing Mechanism
R = Reducer or Concentrator Shaft and Parts
S = Shifting Part of Carriage
T = Typewriter Parts

SUBDIVISION OF EACH SECTION

0–9 Shafts
10–29 Studs, Pins and Screws
30–44 Collars (Screw Machine Parts having Holes)
45–49 Castings
50–74 Punched Parts
75–79 Drop Forgings
80–89 Springs
90–99 Miscellaneous Parts not in above classification

A general system was also required in the designing of the machine itself in addition to the numbering of the individual parts. Iron castings were determined upon to form the bases and the main frames on which the other parts are mounted. Most of the latter are made of sheet-metal punchings, combined with and suspended on screw machine parts, and so mounted on the iron castings. Each section of the machine was designed to be, so far as possible, a complete working unit, interchangeable on different machines. Each section was arranged to have the fewest possible points of connection to the operating means, since it is easier to obtain synchronism where the motive power is derived from a single source rather than from a plurality of sources.

25 Next there was the grouping of members and parts for similar functions. This grouping is of twofold advantage. In some cases it is of material benefit to the operator of the machine, and in other cases of benefit to the assembler in putting the machine together. The necessity for this grouping and segregation of parts is almost as great in a machine of this kind as in a department store, where it is easy to imagine the confusion that would result if the store were maintained and all the departments abolished. In consequence of this grouping the typewriter keys are entirely separate from the adding machine keys, the operation keys of the adding machine are in a single column on the left-hand side of the adding machine keyboard, the adding

machine tabulating key on the adding machine keyboard, and the typewriter tabulating key is with the typewriter keys.

26 Regarding the inside of the machine it can be said that all shafts which have a constant motion when the handle of the adding machine is operated are controlled by the mechanism on the right-hand side of the machine, while shafts which have an irregular movement in relation to the handle movement are controlled by the mechanism on the left-hand side of the machine.

27 The adding machine itself is mounted on two vertical cast-iron side frames extending longitudinally of the machine. The parts between the two frames are primarily for executing the work of calculation and printing, while the parts on the outside of the two frames are for determining and directing the operation of the parts between the frames.

28 These facts may not seem to have much bearing on the problem of assembling the machine itself, but they are of extreme importance in this connection. By knowing the location of the parts which do the work the assembler is assisted in inspecting the work as he goes along and is instantly enabled to locate troubles and faults. If the machine were designed without this separation of functions it would be very difficult indeed to determine just what the trouble might be in many cases where the parts are out of sight.

29 There are many other limitations in the designing of a machine of this character, such as the convenience of the machine to the operator, so as to enable him to attain speed in the execution of his work; the character of the parts and the method of producing them, such as punchings, screw machine parts, castings, etc., and the means of machining and finishing these parts so that they can be used interchangeably. Esthetic considerations must be applied to determine the size, form, color and finish of surface, ornamentation, etc., of the machine as a whole. Also, the assembling and operation requirements call for consideration in the design. These include the limitations of the materials in strength, lightness, hardness, elasticity, etc.; the allowance of clearance to correct errors due to distortion; the spring of parts; the effect of rebound and means of overcoming it; irregularities in castings and in the machining of parts; changes due to variation in temperature, etc. These are some of the points which, if carefully considered by the designer, help somewhat in the ultimate assembling of the machine, and if neglected arise spectre-like to worry the maker afterwards.

THE PRODUCTION OF PARTS

30 System in designing is the first step in the assembling; the second step is the accurate production of individual parts and the accurate assembling of small groups of these parts. In general, the parts in this machine are of three kinds: punchings, screw machine parts, and castings. The latter have various machine operations on them.

31 The punchings of which this machine is mainly composed, are all produced from dies similar to that shown in Fig. 8, and

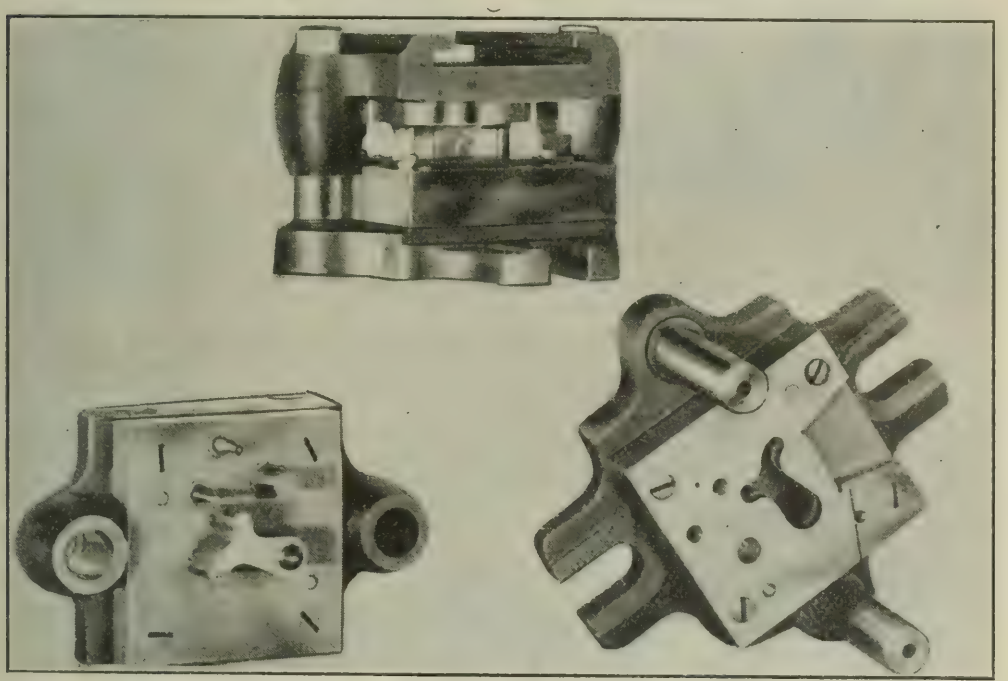


FIG. 8 TYPICAL BLANKING DIE

are, properly speaking, sub-press dies. The work, however, is not returned into the stock as in many forms of sub-press work, but passes through the orifice in the bottom of the lower bolster. Over 400 dies of this general character are required, and the component parts were made of standard sizes and in considerable quantities. The bolsters and caps were planed or milled in good-sized lots and then drilled and reamed for the guide posts, which were ground to size in lots. The stock parts for the dies, strippers and punch holders were also made up in quantities of standard size, carefully ground top and bottom, and the die stock was labeled with the brand of steel and blued for the layout. There were, of course, a number of standard sizes of stock sub-

presses. The punch stock was not made in standard sizes but was cut specially in each case.

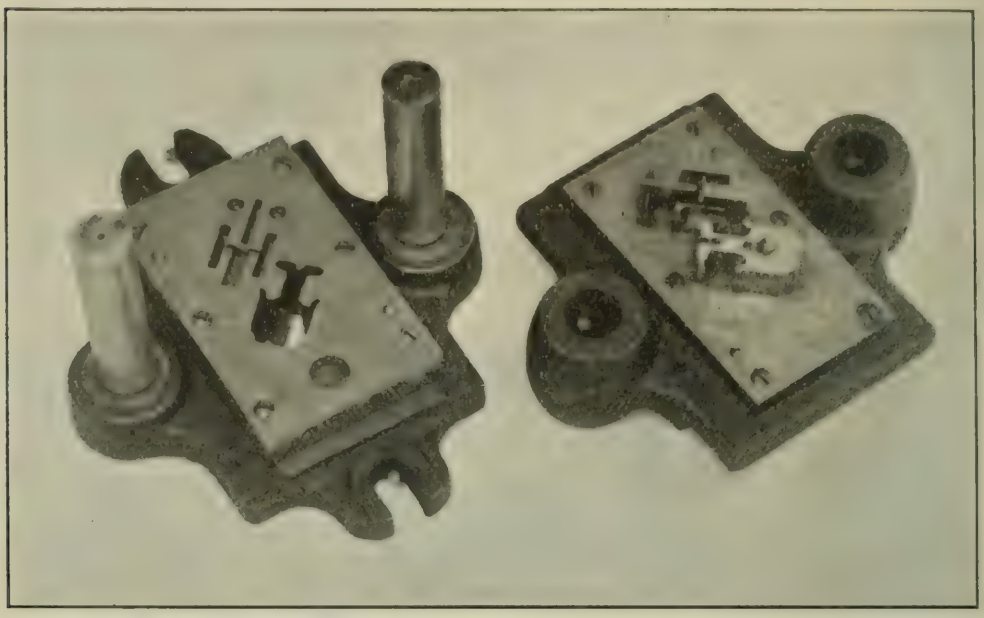


FIG. 9 TWO-STEP PROGRESSIVE BLANKING DIE FOR 50 I

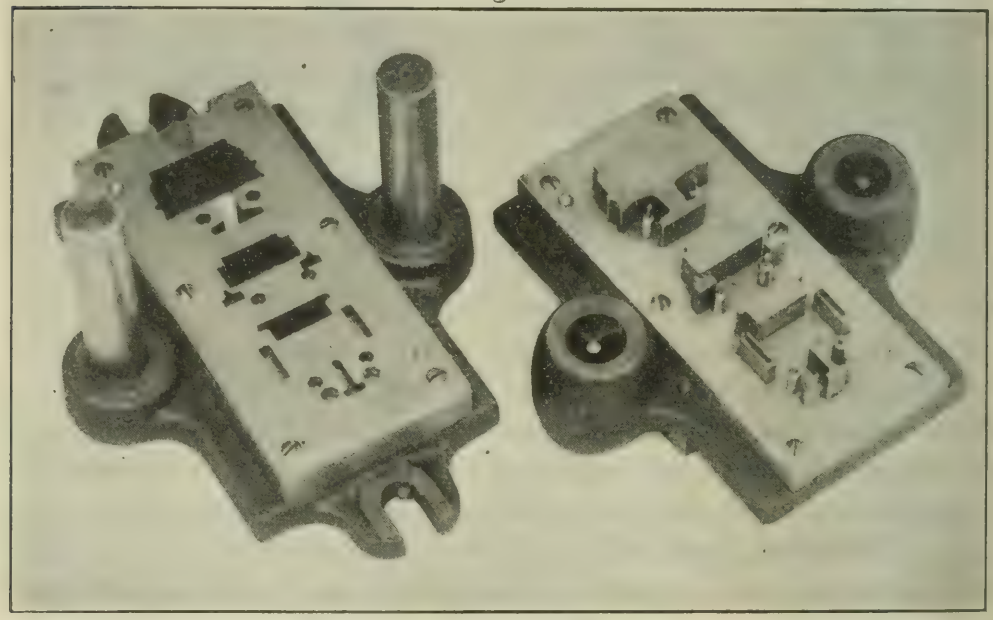


FIG. 10 THREE-STEP PROGRESSIVE BLANKING DIE FOR 51 I

32 While the great majority of punching dies are of this simple kind, there are also many others much more complicated, as shown in

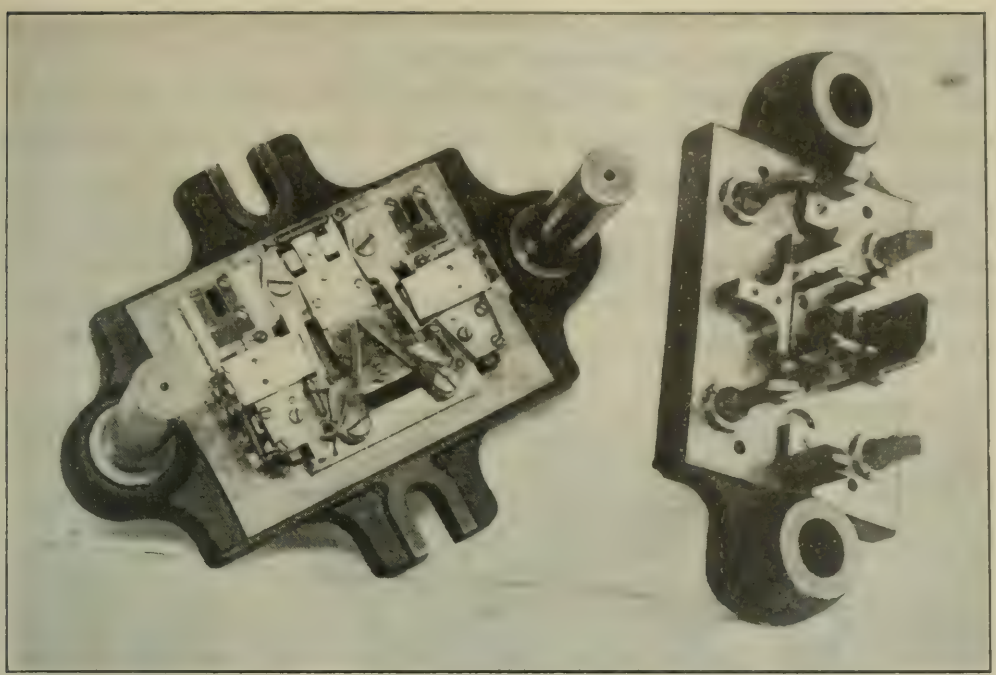


FIG. 11 FORMING DIE FOR 50 I

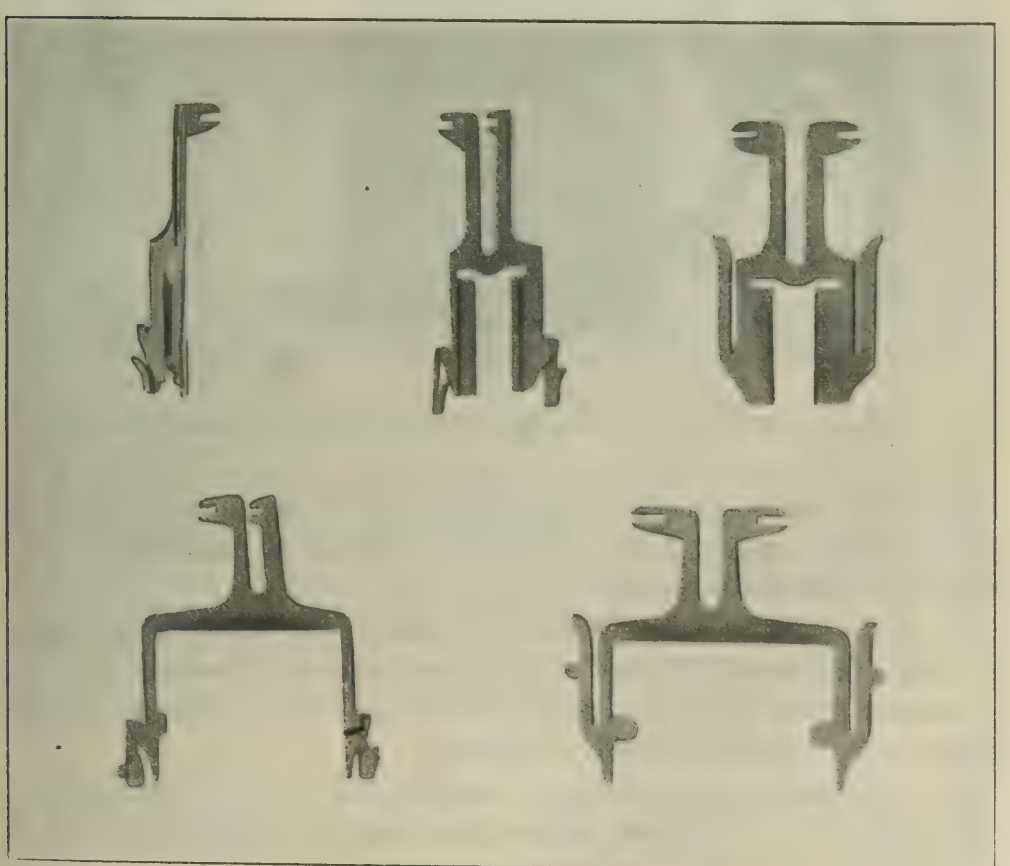


FIG. 12 RIBBON VIBRATORS 51 I AND 50 I

Fig. 9, which is a two-step progressive blanking die, and in Fig. 10, which is a three-step progressive blanking die.

33 Bending, drawing and forming dies are mounted, in most cases, in the same type of sub-press and vary in character from the most simple to quite complicated ones. One of the more complicated dies is shown in Fig. 11, and the piece it produces, in the flat and finished shape, in Fig. 12. The flat blanks being produced in the dies shown in Figs. 9 and 10 are the ribbon vibrators, which are entirely bent to the finished shape by one operation of the bending die.

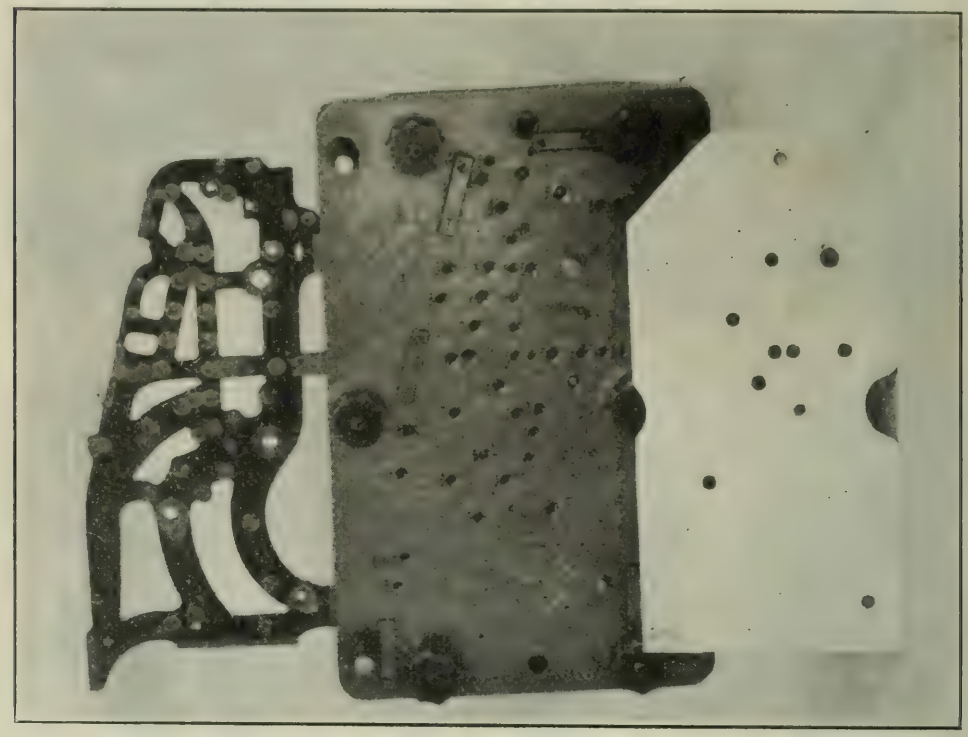


FIG. 13 LEFT SIDE FRAME B 47, DRILL JIG AND MASK PLATE

34 The making of the dies, together with the other tools for this machine, required the services of about 40 toolmakers for a period of over three years.

35 A few of the screw machine parts are finished on turret forming machines, but most of them are made on Brown and Sharpe automatic screw machines, the processes and tools being in no way different from those generally employed.

36 The castings are the most complicated pieces in the machine, but an effort has been made to keep the machine operations on them as simple and straight as possible. As an illustration, the side frames

Lot. No. _____ For _____ Machines

SYMBOL	No. of Operations	OPERATIONS																				FINISHED	SYMBOL	
		1st		2nd		3rd		4th		5th		6th		7th		8th		9th		10th				
		S	F	S	F	S	F	S	F	S	F	S	F	S	F	S	F	S	F	S	F			

FIG. 14 OPERATION DOCKET

The sheet actually used provides for twenty operations

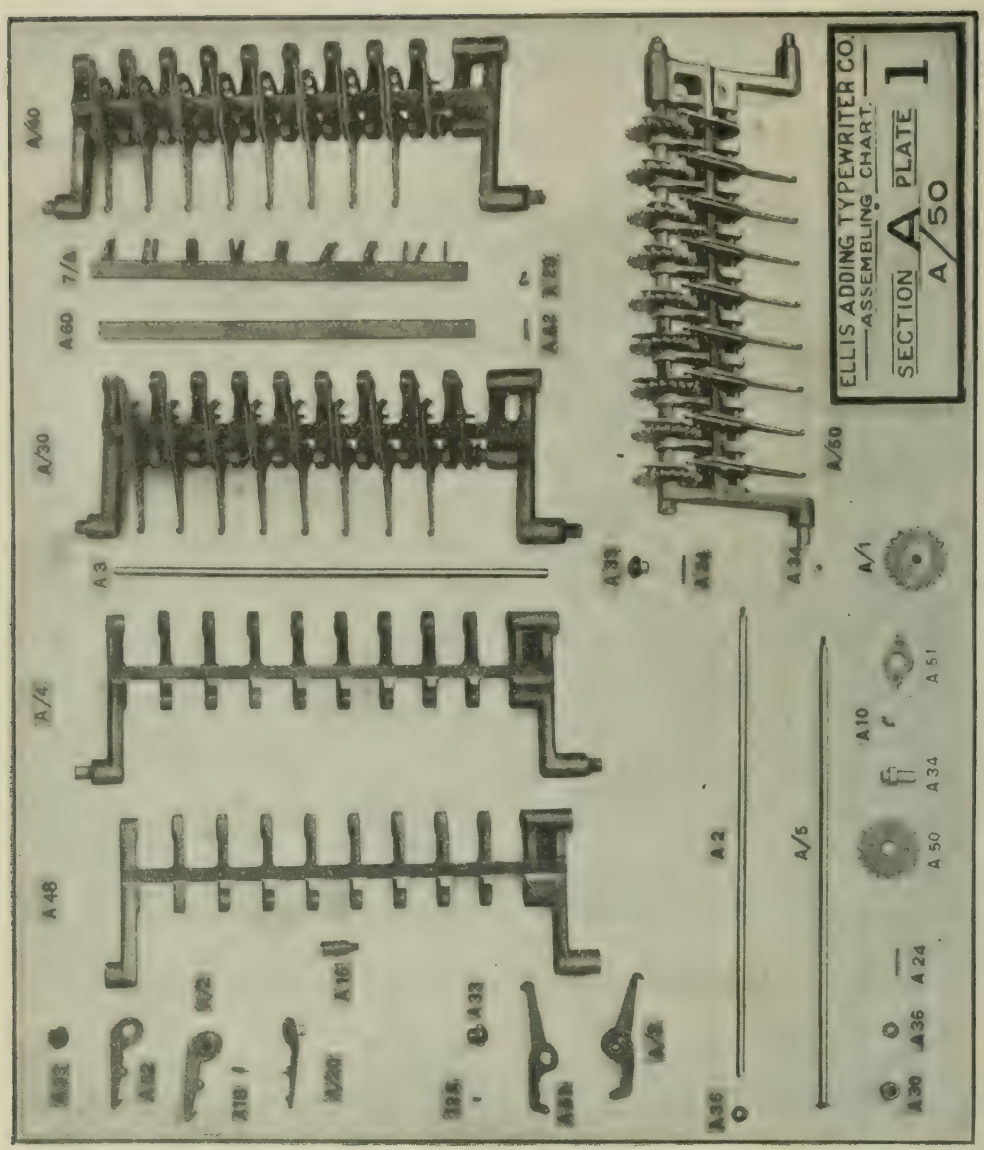


FIG. 15 ASSEMBLING CHART FOR SUB-SECTION A/50

B 46 and B 47 (see Fig. 13) are of uniform thickness through all flanges and bosses. These castings have an allowance of only 1 mm. in thickness for finish on the two sides. They are straightened in a press, after being heated, and are then japanned. Following this they are ground to size on Pratt and Whitney vertical surface grinding machines at the rate of about five minutes per pair. No difficulty is experienced in bringing the two sides of these frames perfectly flat and parallel, with a variation in thickness not over 0.01 mm. After the frames are ground, they are drilled and milled. The left-hand frame, B 47, has 61 holes through it, requiring over 200 operations. The master plate from which the drill jig was made required about 70 days' time of the most expert workmen. The resultant plate contains 66 holes true to drawing within 0.01 mm. As an illustration of the effect of this kind of work on a man, it might be mentioned that after about 30 days' work it was discovered that the workman had made a mistake in his calculations which put about 20 of the holes 0.1 mm. wrong, and the discovery, made by the man himself, so unnerved him that he broke down and wept and had to stop work for the day to recover himself.

37 Stock records of the manufactured parts of this machine are kept on typical card forms adapted to the peculiar requirements of this machine. The operation card, perpetual inventory, and stock order forms are too well known to be shown; but the form called the docket (Fig. 14), which shows the progress of the work on every piece during the progress of a lot of machines, may be of interest. By turning over the pages of this docket, it can be seen at a glance what work has been done and what remains to be done on every individual piece composing the machine; how long it has taken to do the work; and, by referring to back entries, how long it will take to finish the work. This form is much more visible than any card index system, and is correspondingly more useful to the foremen and superintendents.

ASSEMBLING THE SECTIONS OF THE MACHINE

38 The assembling of the machines is first done by building up what are known as sections, after which the sections are connected to form a complete machine. The growth of a single section will be illustrated by showing how the parts in the A section come together. This section is located at the rear of the machine. It consists of two entirely separate and independent adding mechanisms

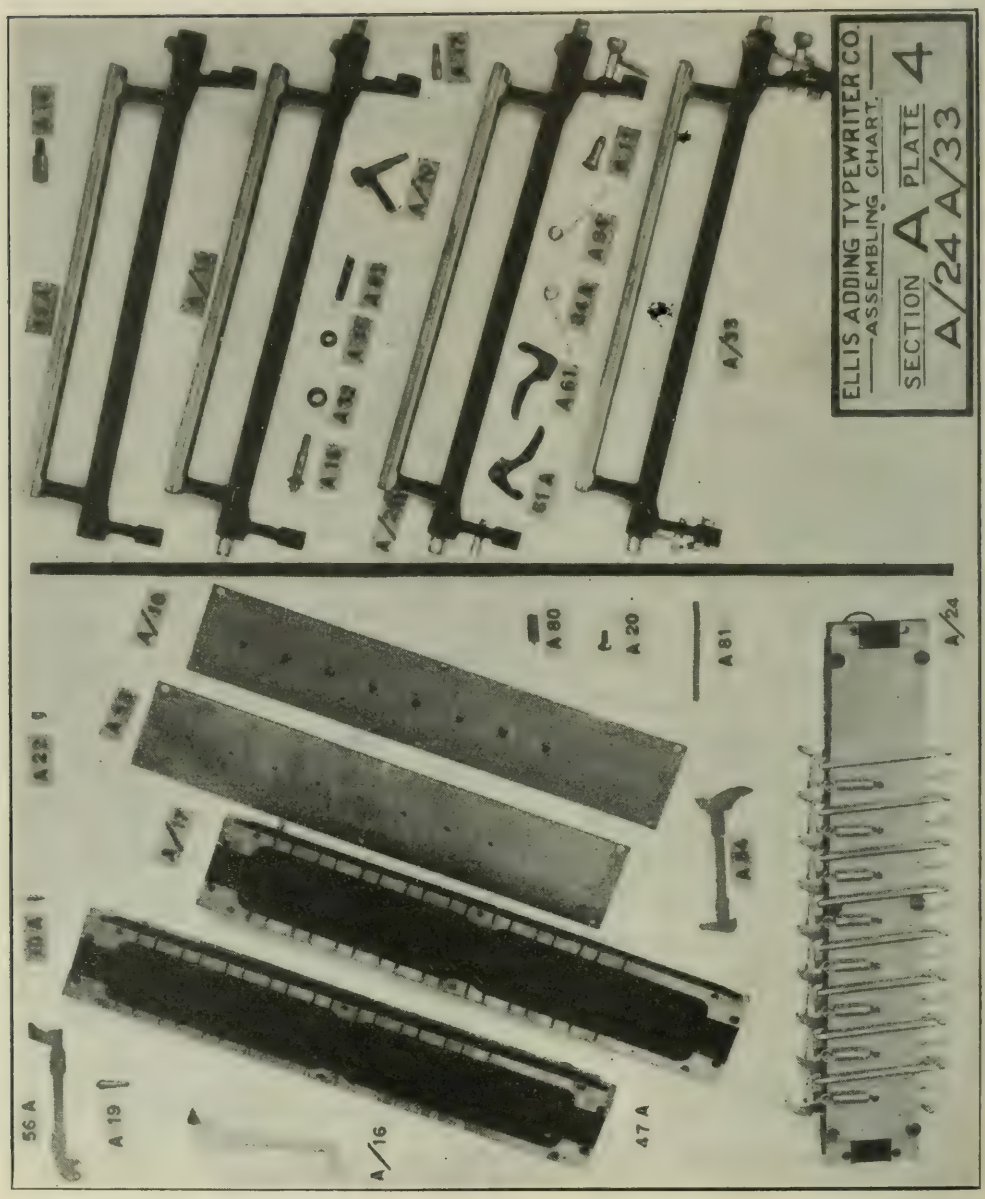


FIG. 16. ASSEMBLING CHART FOR SUB-SECTIONS A/24 AND A/33

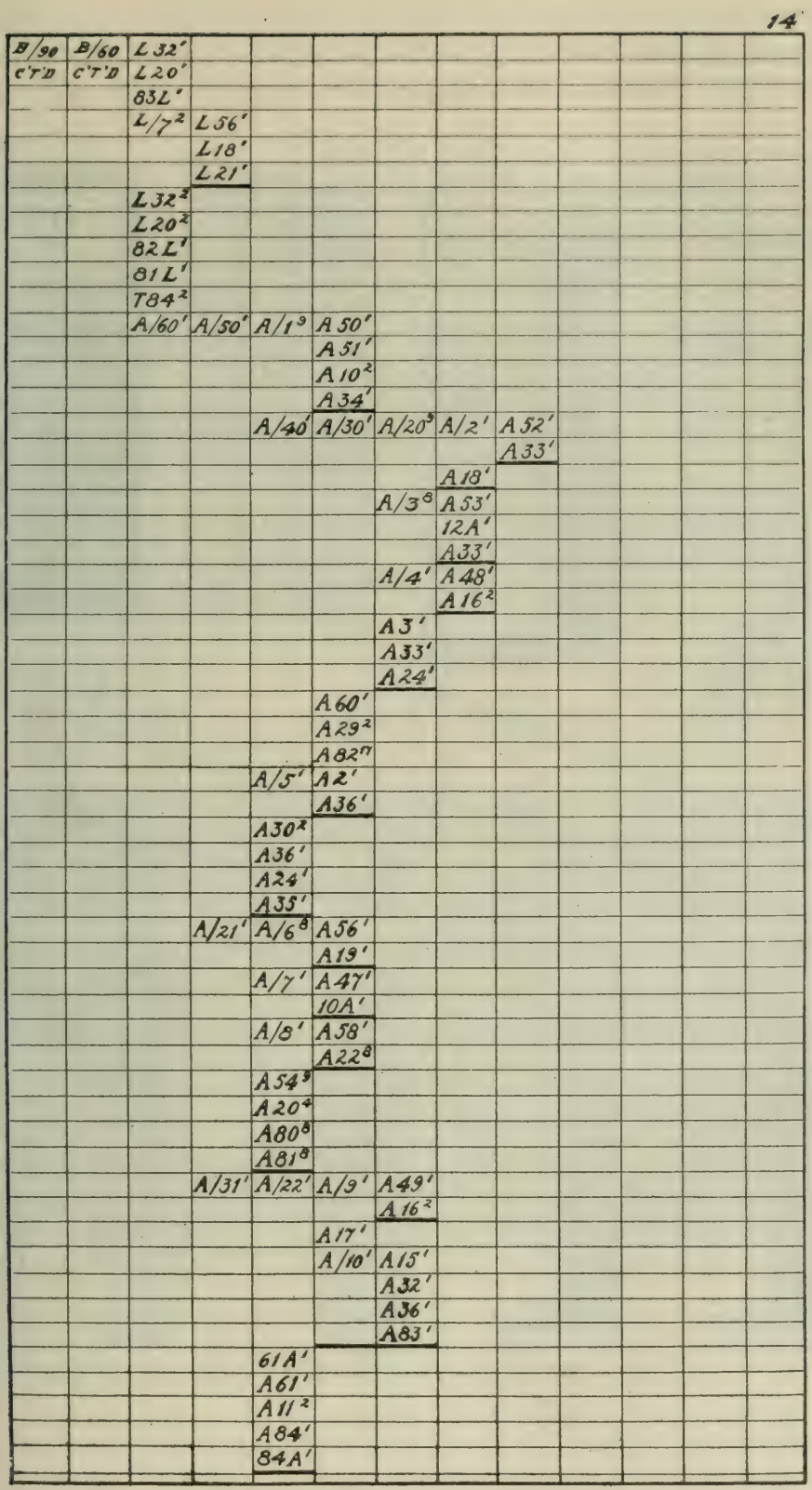


FIG. 17 TYPICAL SHEET FROM ASSEMBLING CHART

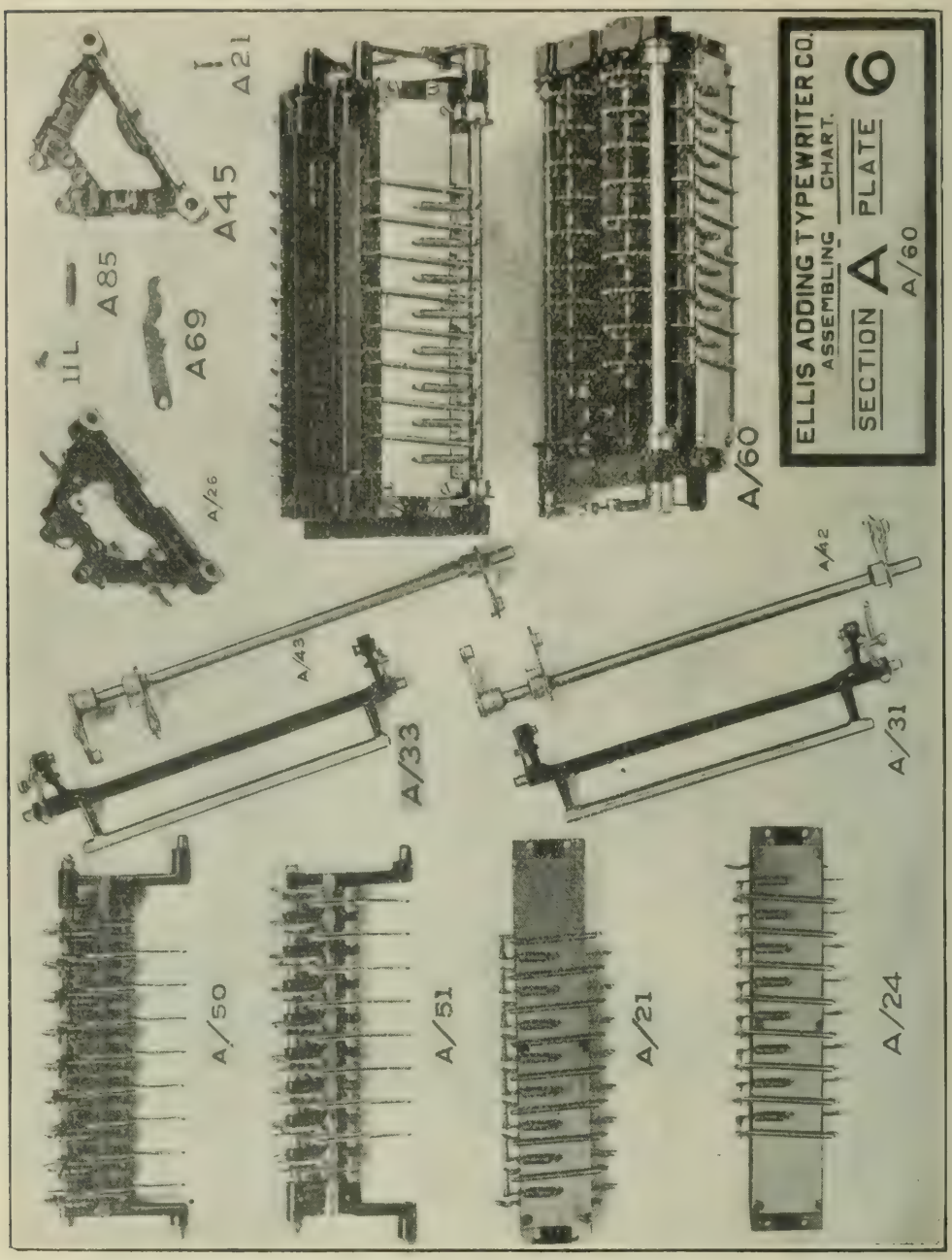


FIG. 18. ASSEMBLING CHART FOR SUB-SECTION A/60

mounted in a single frame, each being controlled by a single connecting bar from the operating mechanism of the machine, as shown in Fig. 7. The individual parts that compose section A are assembled as

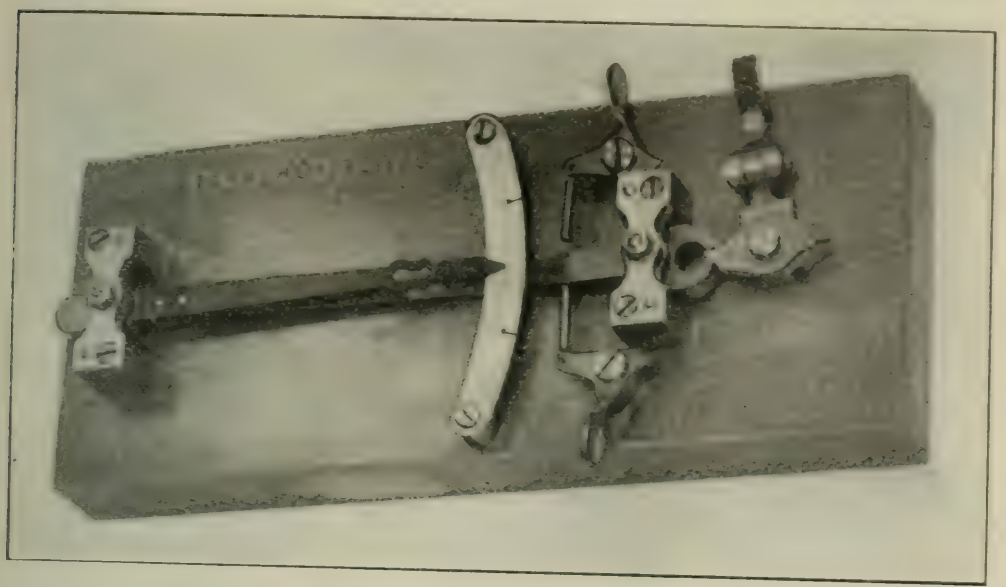


FIG. 19 TEST GAGE FOR A 51 AND 51 A

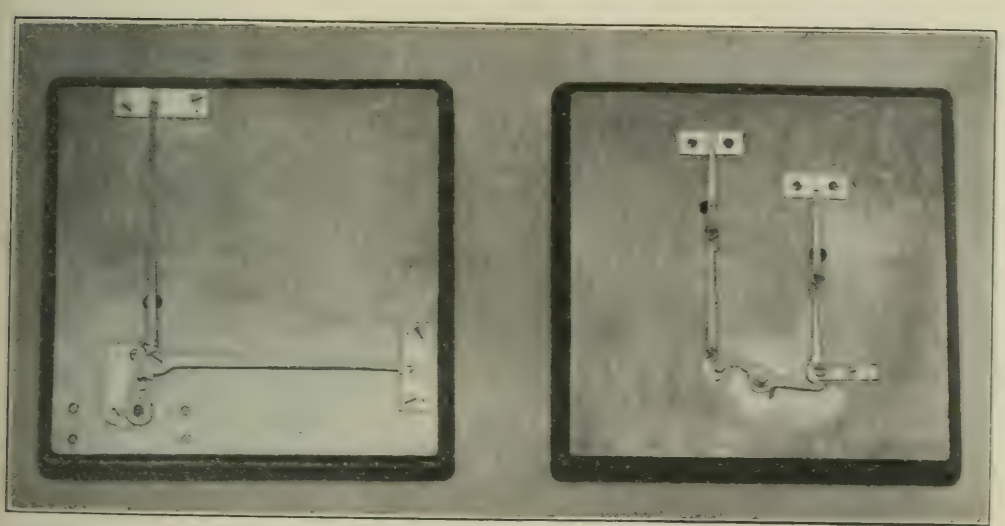


FIG. 20 TYPICAL TEST GAGES

shown in Fig. 15. Here a series of parts are shown in progressive steps, from the original machine pieces up to sub-section A/50.

39 A 33 and A 52 combine to make A/2, to which is added A 18, making A/20. In the same way 12 A, A 33 and A 53 are assembled, making A/3. Two screws, A 16, are inserted in the ends of A 48, mak-

ing A/4. The shaft A 3 is inserted through a series of holes in A/4, and as it is inserted eight pieces A/3 and nine pieces A /20 are strung

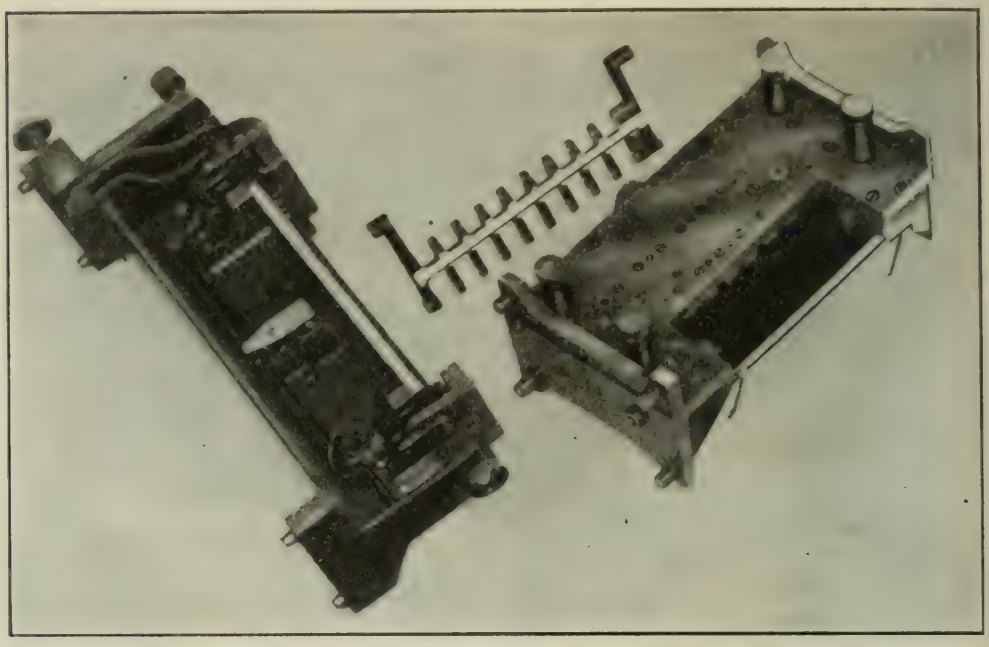


FIG. 21 ASSEMBLING DRILL JIG AND DRILL JIG FOR A 48 AND 48 A

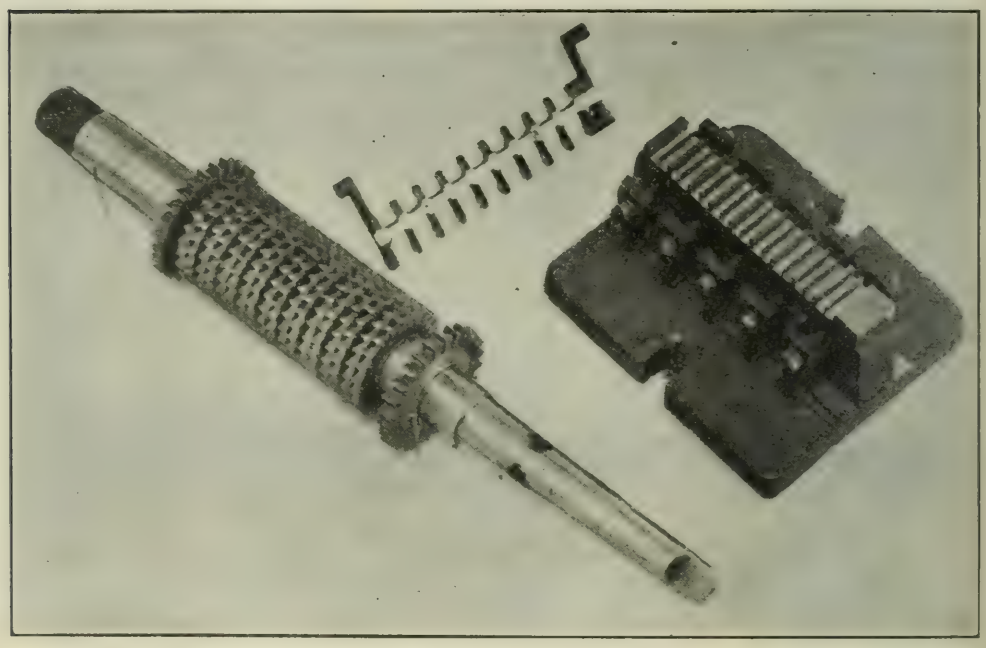


FIG. 22 MILLING FIXTURES AND GANG CUTTERS FOR A 48 AND 48 A

upon it, making A/30. Next a perforated strip A 60 has a series of springs A 82 connected to it, making 7/A, which is mounted on A 30

by screws A 29, and the springs A 82 are connected at their free ends to the small pins 12 A of the pieces A/3. There are nine pieces A/20,

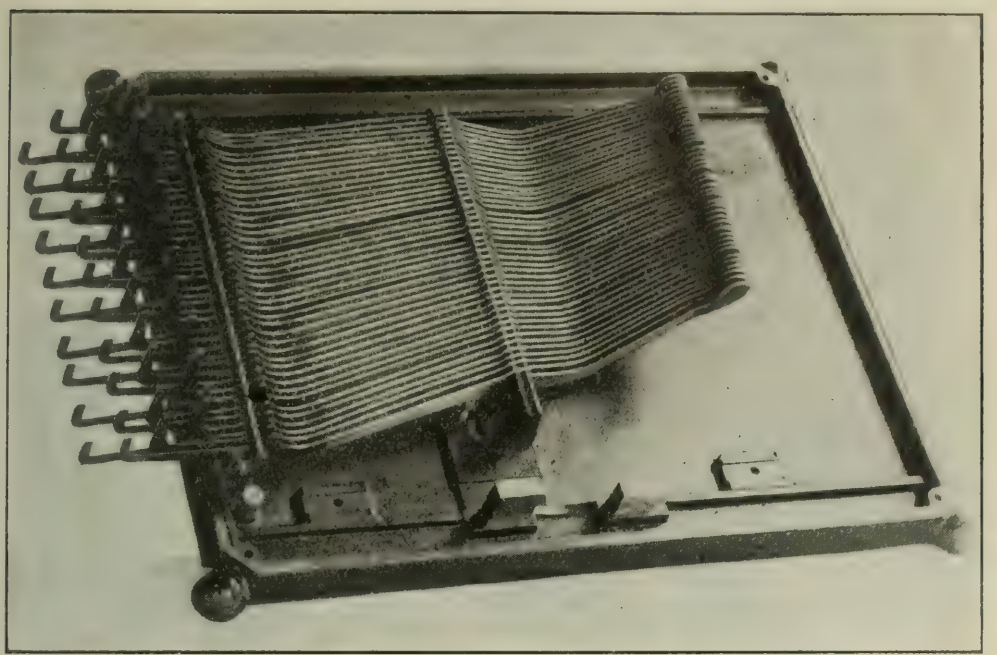


FIG. 23 SECTION B/30

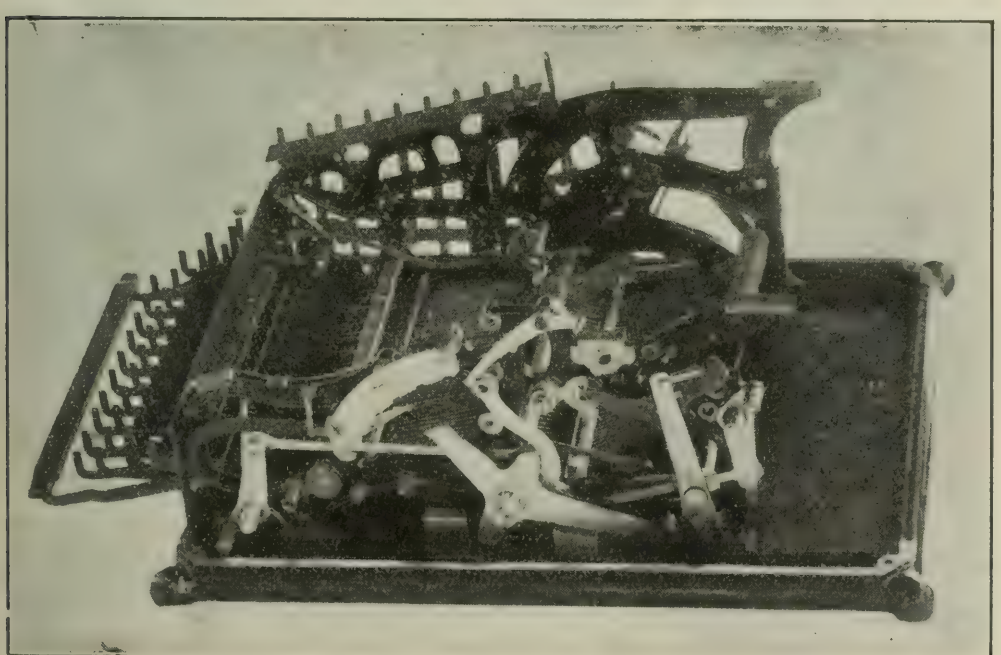


FIG. 24 SECTION B/50—FIRST OPERATION

employed and to fill up the gap which would otherwise be occupied by the piece A/3, an extra hub A 33 is inserted in the space between

the piece A/20 and the side of A 48. In the same way the adding wheels A 50, their side carrying cams A 51, are mounted on the hubs A 34 and secured by rivets A 10, forming a piece A/1, and a suitable bearing rod A/5 serves as a pivot about which these wheels A/1 can revolve, shaft A/5 being inserted through holes in the end of the piece A/40. When these wheels are mounted on the shaft A/5, in the section A/40, the piece is complete and becomes A/50. This is known as a sub-section, which is a complete unit in itself, but goes to form a main section.

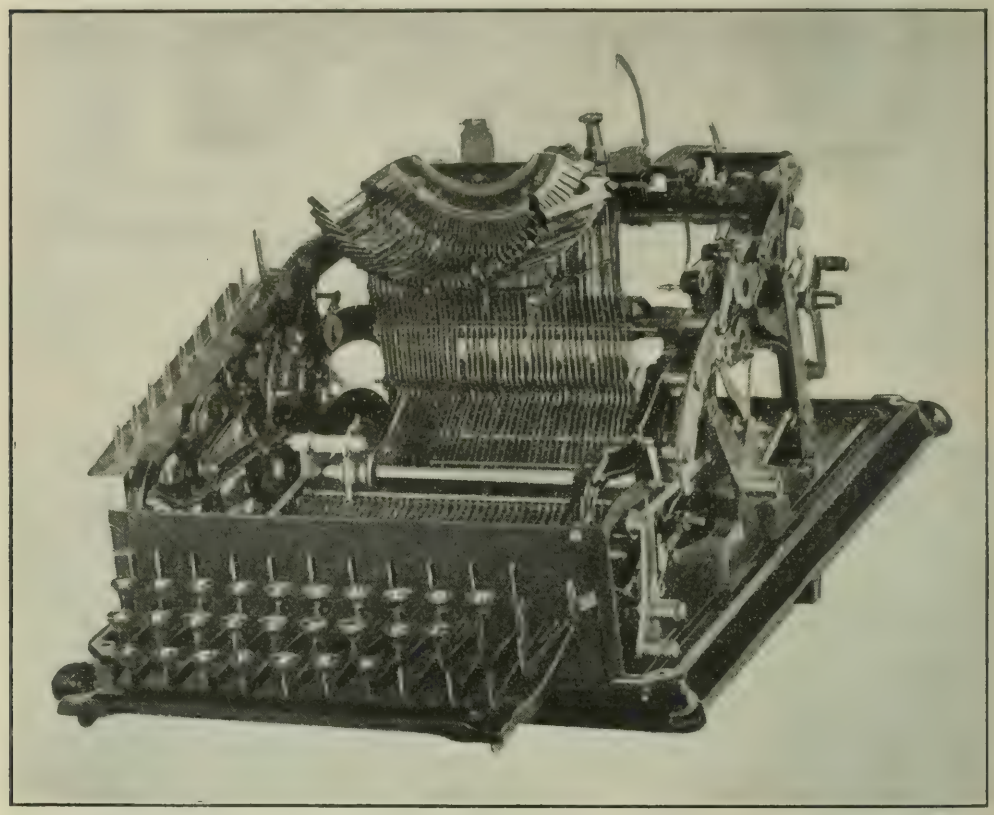


FIG. 25 SECTION B/50—SECOND OPERATION

40 There are two complete and independent adding devices in this machine, made right-handed and left-handed, not in all respects identical. Fig. 16 shows the parts for sub-sections A/24 and A/33, to be assembled left-handed, and the progressive stages in assembling.

41 This method of illustrating the different steps in the assembling and the use of charts (Fig. 17) to simplify the explanation to the workmen is found very useful. It has not come within the writer's experience to find a man capable of carrying this mass of detail in his head

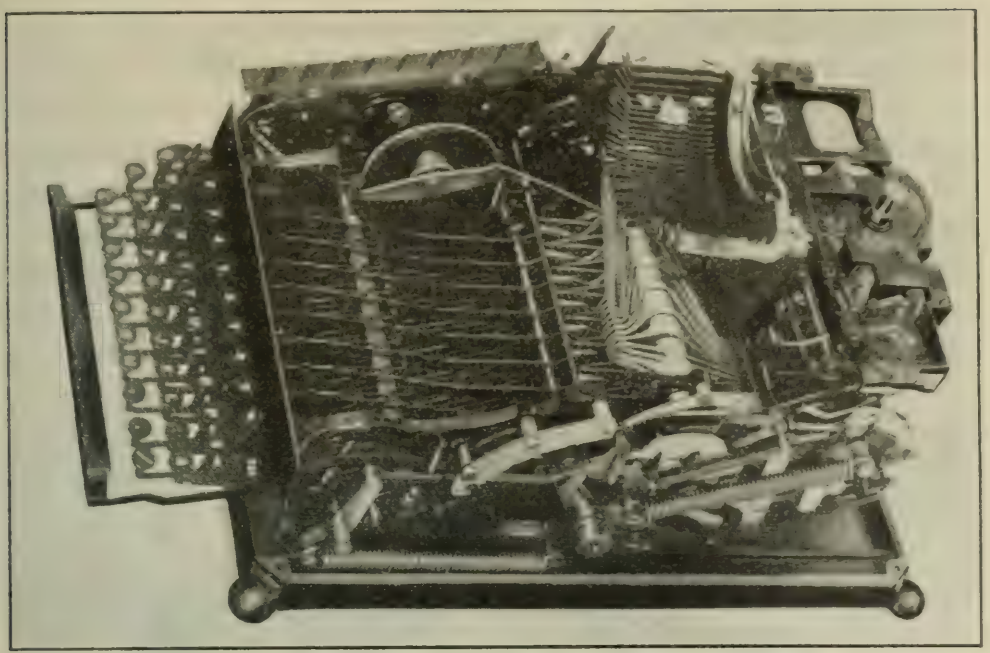


FIG. 26 SECTION B/50—THIRD OPERATION

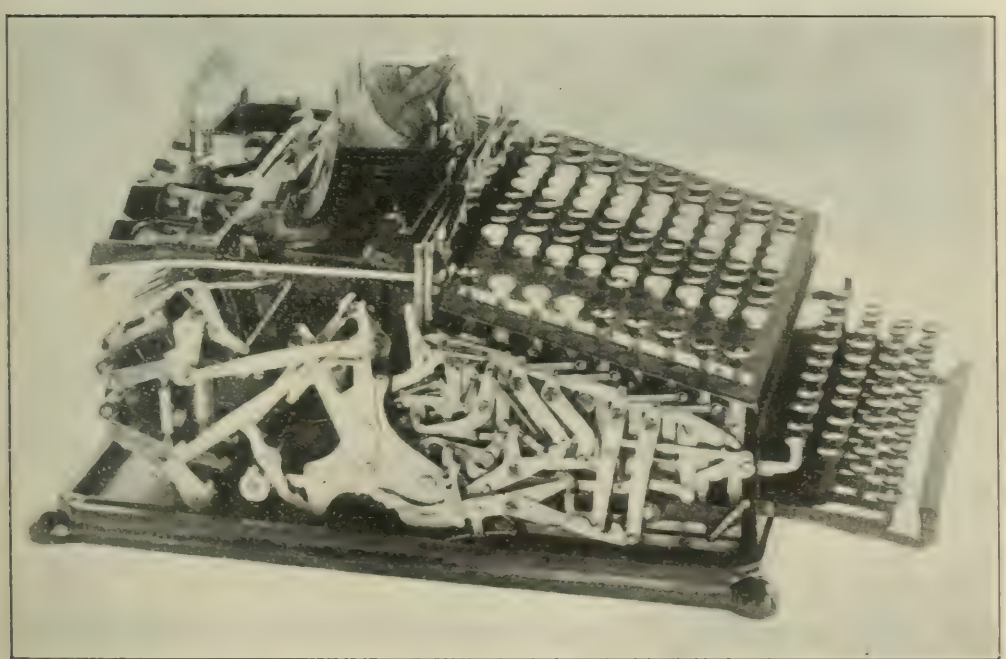


FIG. 27 SECTION B/60—VIEW OF LEFT-HAND SIDE

unless aided by the best memorandum system that can be devised. Of such methods the photographic chart system is the most comprehensive that the writer has so far considered.

42 After the sub-sections are assembled, they are gathered together as shown in Fig. 18, and this forms a complete working unit known as a section. This unit is interchangeable on different machines and can be detached by the removal of six screws. The other individual sections of this machine are assembled in the same general way.

43 Some of the details of the production of the individual pieces, and the assembling of these pieces are shown. The testing gage for the pieces A 51 and 51 A is shown in Fig. 19. This is a compound-lever testing gage which magnifies the reading 100 times, and the error is 0.01 of the space indicated on the dial by the pointer. Other

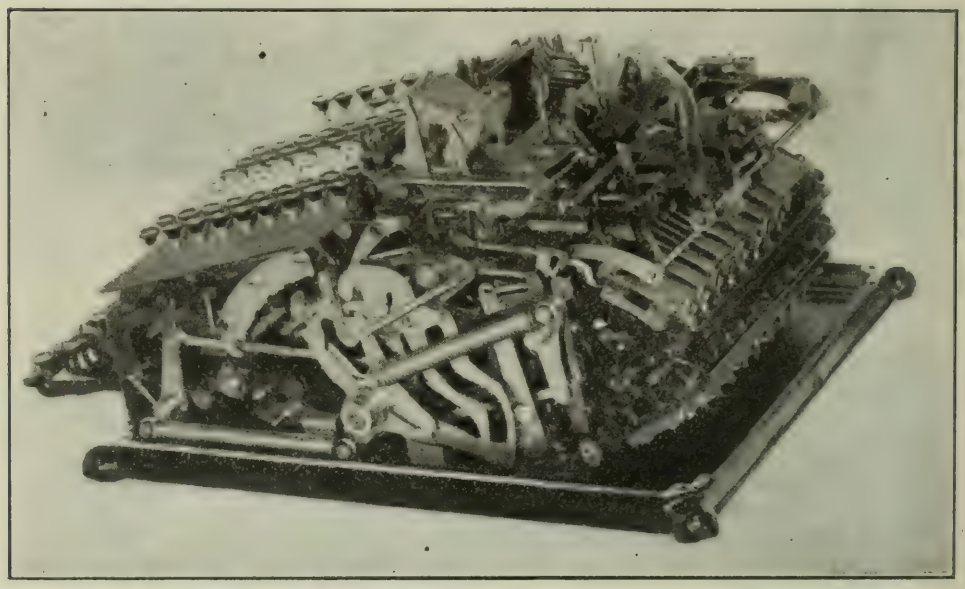


FIG. 28 SECTION B/60—VIEW OF RIGHT HAND SIDE

test gages, simple and compound, are shown in Fig. 20. In Fig. 21 is shown an assembling drill jig, and a drill jig for the pieces 48 A. In Fig. 22 is shown a milling fixture for the piece A 48 and 48 A and a typical gang mill.

ASSEMBLING THE COMPLETED MACHINE

44 The assembling of a completed machine starts with the base, on which are mounted the typewriter key levers, as shown in Fig. 23. The next step is to mount the two side frames B 46 and B 47 thereon, after which the parts between the side frames are inserted, and connected to the side frames, Fig. 24. The escapement, known as the E/60 section, is next mounted on the side frames, the typewriter connec-

tions are put in, and the machine assumes the appearance indicated in Fig. 25. The next step in assembling is the insertion of the R section or parts, consisting of the racks, the reducer arms, and the governor, all of which are shown in Fig. 26. The next two sections to be inserted are the F/70 section, and the P/70 or printing section of the adding machine. There remain the K and A sections, which are easily and quickly connected. The machine then assumes the appearance shown in Figs. 27 and 28, and is ready to have the case fitted over it and to have the carriage mounted upon it.

45 It may sound rather platitudinous to remark that the mechanical and commercial possibility of the manufacture of a machine of this kind resolves itself into a question of system; not so much a system in the putting of the parts together as the system required in designing the machine in the first place so that the assembling can be properly subdivided and each section tested for its ultimate fitness in the complete machine.

No. 1312 *c*

QUANTITY MANUFACTURE OF SMALL DEVICES

By F. P. Cox,[1] West Lynn, Mass.

Non-Member

46 The economical manufacture of small devices in large quantities is chiefly characterized by the attention which must be given to details. The cost of material is in general proportionally small, and in any case more or less fixed by the design; but the labor item is large and requires constant and careful scrutiny. Unlike other lines of manufacture, each operation is repeated over and over again and, therefore, warrants a considerable expenditure of time and money to determine exactly the best manner in which to perform it. It is desirable to know just how economically it can be performed even though temporary conditions prevent an immediate realization of possible results.

47 This leads to scientific rate setting, and the advantages of such an experimental manufacturing department are too well known to require comment. It means automatic machinery for many operations, but not for all; it means suitable provision for stock handling, inspection, supervision, etc. Each of these subjects, if covered completely, would require an individual paper. In addition, it requires organization to provide for the proper correlation of these various elements and to insure their harmonious operation.

48 Many plans of organization along the same general lines are possible. The general lines are subject to variations at times, but they are all centered around the same principal object, the labor item. No manager will ever be able fully to realize his ideal system, and perhaps the chief reason will be found in the limitations of his assistants. An organization which has been in operation for any length

[1] Meter Dept., General Electric Co.

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

of time will have developed a certain number of individualities which must be reckoned with in any contemplated change. It would be folly to dispense with a man of proved value because he does not happen to fit exactly in an ideal plan of organization. It is always possible to modify the system to avoid such a sacrifice and without any material departure from the original scheme.

49 An equivalent system can be devised with comparatively little difficulty, but the equivalent of a tried man is hard to find. Of course this does not mean that a weak man should be retained, nor that one not in sympathy with the plan of organization should be permitted to take part in its administration. Neither does it mean that any effort should be spared to find and employ a competent man to fill an important part in the general scheme. It does mean, however, that a man whose known limitations prevent him from filling all the duties of a certain position should have those duties so modified that he will not be prevented from filling those for which he has been found particularly adapted; and that one who has shown particular ability in handling two or more lines, not originally intended to be combined, may still exercise this talent by a reasonable modification of the system.

50 The organization here outlined probably differs very little from similar organizations intended to cover the same field.

Manager
- *a* Superintendent
  - (1) Production clerk
  - (2) Stock clerk
  - (3) Material list department
  - (4) Rate setting department
  - (5) Automatic tool department
  - (6) Shop accountants
  - (7) General foreman
  - (8) Inspectors
  - (9) Tool department
- *b* Engineer
  - (1) Designer
  - (2) Draftsmen
  - (3) Model room
- *c* Cost department

51 The strength of this organization lies in the fact that each feature of manufacture is under control of a man particularly qualified

to handle it in the best possible manner, and that his responsibility is definitely fixed and limited to the proper performance of his own part of the work. There is no overlapping of responsibility and no possibility of misunderstanding. Any failure to produce a part is at once traced to the responsible party. For example, if a leading hand finds that he is not receiving his parts in adequate numbers and is, consequently, handicapped, he at once reports to the stock clerk who has failed in his duty. The stock clerk realizes that he has failed. The mere absence of parts is convincing evidence of the fact, and he must correct the shortage before it becomes a limiting feature or report his inability to do so to the superintendent.

52 In this way, the superintendent is constantly advised of matters which are liable to interfere with his business, and is able to apply corrective measures before the situation is serious.

53 When everything is moving properly, if such a condition should ever exist, the work of the various departments becomes routine, but the moment any trouble appears in the distance, an alarm is sounded. If trivial in character, it is corrected without any great disturbance, but if important, it comes at once to the highest authority and the whole strength of the organization is applied promptly to correct it.

54 The functions of the different departments are, in general, sufficiently described by their names. It may be necessary to add that all material is ordered by the material list department, but the following of the stock into and through the shop rests with the stock clerks, and that while the inspectors come under the superintendent, the limits of inspection are set by the engineers.

55 The workman is the principal subject of study and every effort must be made to shorten the time and lessen the effort required to perform the operation on which he is engaged. It may mean automatic tools, increased facilities, improved methods, or the splitting of the operation so that a portion of it may be done by a less skilled operator, who is competent to do this portion in a satisfactory manner.

56 Always inspection is involved in order that no time may be spent on a piece which is already defective and will ultimately be rejected or require special fitting, which is necessarily expensive. The proper time to reject a piece or to correct it is immediately after the false operation has been completed.

57 In order that a workman may perform a maximum of work with a minimum of effort, it is necessary that the piece shall come to him in ample quantity and at a constant rate; that previous operations shall have been properly performed, and that the piece shall be

promptly inspected and removed. His whole thought and effort should be given to performing his own operation satisfactorily. One portion of this desired result may be obtained by a large stock of parts, but this means unnecessarily large inventory, and should be rejected. It is just as necessary that parts should not wait for the workman as it is that the workman should not wait for parts.

58 Regular and frequent meetings are held by the superintendent and his assistants, so that each may be familiar in a general way with conditions as a whole, but such meetings are not to be construed as lessening the responsibility of the different departments. These meetings require but little time if everything is going smoothly, and justify the time taken if there is any prospect of trouble.

59 Similar meetings are held weekly by the manager, engineer and superintendent, and are more or less generally attended by principal assistants. Changes of design or method are freely discussed at these meetings, and they serve as a clearing house for all matters of interest which have come up during the week and which are not of sufficient importance to require instant attention.

60 Changes of design, unless to correct defects in the apparatus are docketed and not acted on singly as they arise. Low costs cannot be obtained where the design is continually changing, and minor changes should be held for an apparatus meeting. Such meetings are held at regular intervals of from three to six months. When a line of apparatus has but recently been placed in the shop, the interval is shorter than when it has become more firmly established and original troubles eradicated.

61 Apparatus meetings are attended by the manager, engineer, cost department and superintendent, with their assistants who have the actual handling of the apparatus. The device is placed on the table and gradually dissected, each part being fully discussed as it is taken off, and the presence of the leading hand is considered essential during the discussion of the particular part or operation with which he is concerned. The leading hand, or gang boss, comes under the general foreman. He has direct charge of a rather small number of operators, and often a portion of his time is given to productive labor. More than anyone else he appreciates inconveniences and difficulties which arise in producing his particular part, and any organization which fails to profit by such knowledge is fundamentally defective. Introduction of automatic tools, change of methods or of design, and quality of product are thoroughly gone into and after this meeting all changes decided upon are authorized at one time after which none are made until the next change period.

DISCUSSION ON INTERCHANGEABLE PARTS

HUGO DIEMER said that while on a visit to the plant of the Western Electric Company several years ago, he was shown an interesting feature of their system of assembling. In connection with the scheduling of stock orders, each foreman estimated the time of completion, and after the work was done he was given a percentage rating each month based on the accuracy with which he predetermined the time of completion. Certain foremen with a percentage of 40 at the beginning of this practice were able to reach a standard of nearly 90 per cent. He wished to ask Mr. Puchta, as a representative of the company, whether this was merely a temporary effort, or if it had been continued, and whether an inducement was offered to these foremen to acquire greater efficiency.

EDWARD PUCHTA replied that the plan of making a promise for the completion of each special order was still in effect, but the regular shop schedules were placed monthly, and it was expected that they be completed in the standard time. A record is kept of the estimates on special orders, and at the end of the month a statement is prepared showing how many of these promises each foreman has kept. No especial inducement is offered, except that the placing of one foreman's record against another's brings about competition.

C. W. RIPSCH agreed with the methods stated in the papers for the division and outline of the work for the proper assembling of such machines and discussed the proper methods of fastening punchings of rolled steel made on the screw machine and the fastening of this combination to a shaft. The present method of ring staking and pinning the punchings to hubs is quite often a failure in parts which receive much pound, as the pieces loosen after continued operation. Furthermore, the cost of ring staking, drilling, pinning and lapping off the resulting surface is great whether done in the punching and

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hub separately or in a locating fixture. To remedy this difficulty he had experimented with the method of electrically spot-welding such parts, which serves to fasten the pieces quite securely. A chisel is required to chip apart the weld. The condition of the metal is the same except for the slight blueing and ridge thrown up around the welded point which can easily be lapped off. The main difficulty is to secure a point small enough to withstand the heat for the smaller pieces.

The next operation, fastening this assembled piece to the shaft, is usually done by a dowel pin located either centrally or to one side, or by two pins, one on each side. Owing to the running of the drill and spring due to driving dowels, the pieces will not be interchangeable and will in some cases require broaching or filing. Mr. Ripsch stated that he had failed to find a satisfactory method for this operation which would permit the elimination of the file from the assembler's set of tools, as advocated by Mr. Calder in Par. 17.

A. C. Jackson believed that it would be very helpful if Mr. Calder would supplement his paper by giving data regarding the percentage allowances on different classes of work between what is considered maximum by the rate-setter and the output of the average day worker.

Chas. W. Johnson asked about the methods used by Mr. Calder in paying the demonstrators who did the preliminary work in the rate-setting department before turning the work over to the manufacturing department. He wished to know how their earnings compared with those of regular workers, and whether they really worked at their maximum capacity.

N. W. Perkins, Jr.,[1] representing Mr. Ellis, stated that they had found no better way of doing this work than the methods mentioned by Mr. Ripsch.

W. J. Kaup asked whether it was not true that the best men were chosen as rate-setters and if there should not be some discount made for the average workman.

E. Puchta (written). Mr. Calder's paper is one of exceptional interest, especially to those engaged in the manufacture of apparatus

[1]Engr., Ellis Adding-Typewriter Co., Newark, N. J.

composed of small interchangeable parts, where the assembling is a considerable item of the total cost. This, of course, would vary directly with the simplicity of the design and the requirements placed upon the apparatus to insure its satisfactory operation.

About twelve years ago the Western Electric Company, which manufactures a large number of types of telephone apparatus comprising some 75,000 separate parts, started to analyze manufacturing problems along the lines advanced in this paper. The first step was to produce interchangeable parts, which involved a very careful study of design, as well as of manufacturing methods. Later, attention was directed to the assembly of apparatus, and studies were made to determine the most efficient divisions of labor in assembly and to develop tools and fixtures which would enable the operations to be performed with the greatest facility. Since that time the work has been highly developed and all the elements involved in introducing new designs are thoroughly analyzed and standardized before work is started commercially. The adoption of these methods brought up many interesting organization problems ultimately resulting in radical changes in the organization and some changes in personnel.

It is of interest to note that the organization developed in our shops is very similar to that shown in Mr. Calder's paper. There are, however, a few points which we have worked out differently in our study of the problem. For instance, the supplying of raw material is entirely under the control of the purchasing department. Inasmuch as the interchangeability of parts is directly dependent upon the quality of the raw material, the organization should be so laid out that the shop can control the quality of the raw material by furnishing specifications and by inspecting the product when it is received, to make sure that all manufacturing requirements are fulfilled.

No provision is made in the organization shown for raw material inspection nor for the inspection of tools, which we have found is also of great consequence. Both tool inspection and raw material inspection should be closely associated with the inspection of parts, so that the inspectors in these divisions may be kept well informed in regard to shop conditions. These two departments, as well as the departments inspecting parts and finished apparatus, should come under a common head, who, in turn, should report to the superintendent.

The chart provides that the tool designing department and the testing rates and methods department report to the head of one of the branches of the manufacturing department. Inasmuch as these

departments supervise the work in their respective lines in both branches of the manufacturing organization, it would appear that they should report directly to the man to whom these two branches report.

With an efficient organization for conducting the work there is but little doubt that the fulfilment of the ten conditions outlined in the paper will place the manufacture of apparatus on a highly efficient basis.

In our shops we consider the design of a part with reference to its economical manufacture as well as quick assembly, referred to in condition *d*, of such great importance that we have adopted the practice of analyzing completely and discussing each design with the heads of all departments interested. The design is then modified, if necessary, and the particular features which have been found to be objectionable, from a manufacturing standpoint, eliminated. After the designs are finally determined upon and working drawings completed, each part and each stage of assembly are carefully analyzed by the tool designing and methods department. After the various methods are balanced against each other, the most efficient are chosen and the general design of the tools which will enable the operations to be performed with the greatest facility is decided upon. Upon the completion of the tools, and after they have been inspected and approved for quality of work, they are carefully studied to make sure that they will permit of the operation being performed with the greatest dispatch. If satisfactory in this respect a complete time study is made and the operators are informed of the exact method to be followed in performing each step of the work.

The principal difference between our method and that proposed by Mr. Calder is that our department which corresponds to his testing rates and methods department does not actually run the job in the shop to determine upon time allowances. Our practice is to have the tools turned over to the operators who are to use them commercially, and after they have been instructed and have become familiar with them, the testing rates and methods department makes a careful time study of the job. At this stage the final time allowances or piece-work rates are decided upon, after having carefully compared the rates observed with those previously estimated and with established rates for similar operations.

In Tables 2 to 5 showing the assembly work, operations seem to be included which do not properly belong in such a department. For example, Stage No. 1 in Table 2 calls for a reaming operation on a unit part, which apparently could be better done in some manu-

facturing department, since it is not in reality a part of the assembly work. A number of similar instances occur in these tables, including tapping, reaming and bending operations.

The point made in Par. 17 that files should not be used in assembly, is a very good one, since in prohibiting the use of them the quality of workmanship on the parts has been greatly improved.

CLOSURES[1]

JOHN CALDER. Referring to Mr. Puchta's remarks, which touch in part details somewhat outside the scope of the paper, the purchasing, storing and issuing of raw material are entirely under the control of the purchasing department in the organization illustrated, but the specifications and quality are not.

The chart has been necessarily curtailed in the paper and illustrates especially the general relation of other departments to parts assembly. In particular, the general and departmental inspection system, under the chief inspector and his staff, and the reporting to the superintendent and the tool engineer on the condition of parts and of tools respectively, is not exhibited.

The difference between the system described and that mentioned by Mr. Puchta, where the rate and method department does not actually run the job in the shop, or in its own section of the shop, is a vital one. It is, of course, the author's practice to watch for comparison the performance of ordinary piece-work operatives, but it is the very essence of the system described to have at all times an independent set of facts and figures obtained by a reliable staff. This practically eliminates the delays and the tedious process of guessing involved in turning over untested equipment to see what the shop will do with it.

Even the best of shops will make much greater progress in capacity and arrive quickly at more economical production if it does not start to manufacture a new piece in the dark, with only tradition and custom for guides. The best results actually demonstrated by experts who are far more critical and resourceful than the average employee, should be used from the beginning.

Mr. Puchta's definition of assembly, embracing only work wholly performed by manual operations, is rather a narrow one. Power aids to the assembler have long been used by the author. The com-

[1] Mr. Ellis did not desire to present a closure.—EDITOR.

pounding of unit pieces, as distinct from the subsequent process of assembling machines described by Mr. Ellis in his paper, should not be restricted purely to hand work.

Many slight reaming, tapping and final setting operations can be performed properly to gage only when the finished and related unit pieces are in contact. The latter are often finished in a different manufacturing department from that in which they originated and the provision of some light machine tools in the parts assembly room is not only a desirable, but very economical proceeding. Other handling and transmission delays are thus eliminated and the responsibility for the assembled parts placed solely on the foreman and inspector of that department.

Referring to the request of Mr. Jackson for further details of the time and study system, the main effort is directed towards finding a sure basis on which to determine correct methods and fair piece rates for various classes of task and ability. Each class is rated on a percentage allowance of the study time, based on experience but usually 10 per cent less than the maximum performance. Thus all can make a fair wage and the exceptional men get all their surplus. The latter, however, does not compare with the variations under premium systems.

In the case of fast assembly on very light work by male demonstrators it is sometimes necessary in rating to raise instead of lower the standard performance, as in the case of the greater dexterity of females in work of a light fingering character involving no mechanical skill.

Answering the question of Mr. Kaup, it is not easy to get men to fulfil the functions required from a time and method study department. In fact, there is nothing easy about the task of management, but the author believes that the outside expert is not always the inevitable or the best solution of the matter. Most plants already contain enough men suitable for this work if trouble is taken to locate and develop them. Such men when organized for study work are paid a high daily rate, without reference to what task they may be engaged upon and are really retained for their potentialities. It is possible to determine from the records beyond any doubt whether the demonstrators are giving efficient and consistent service.

The effect of such time and method study work and advance criticism is that, if a body of operatives be given an entirely new job, they know the management is well aware of what can be done and that the employer has carried independent criticism of his own plans

as far as practicable. If there is any doubt in their minds as to whether the rated performance can be accomplished, a demonstration is at once offered and the matter settled. This method is absolutely fair to all concerned and under these conditions the test is seldom demanded.

Confidence in the work of the demonstrators is based on the integrity and reliability of the men selected and on the ability of their supervisor to see that the work done on test is of the prescribed quality and a full measure of their powers.

The author desires to express his agreement with Mr. Cox in placing as much stress upon the development, full use and recognition of able assistants as upon the system which is employed, for the human element is half the problem. At the present time there is a tendency towards too great rigidity in some of the shop systems offered for general application. It is not a recommendation for any system, but rather the reverse, that absolute conformity to type in details without regard to the problem in hand and the great amount of experience already acquired from it are insisted upon. The best shop management is that which will draw out and adequately reward the best effort of everyone concerned, not forgetting the employer, and the best systems for so doing will never be alike in any two cases though the principles may be identical.

F. P. Cox in reply to a question, said that in the conferences mentioned in his paper it was possible to get honest opinions from leading hands quite as well as in personal conversation. The leading hand is solely responsible for his own operations and will frankly discuss them, although as a rule he is more reticent about operations other than his own.

No. 1313

MILLING CUTTERS AND THEIR EFFICIENCY

By A. L. DeLeeuw, Cincinnati, O.

Member of the Society

The amount of metal which a machine tool can remove in a given time is limited by the strains caused by the cut. Great hardness of the material to be cut, or a dull tool, will severely strain the machine and so reduce the section of the chip, even if the machine is rigidly constructed and well supplied with driving power. It is therefore of the greatest importance to analyze carefully all the conditions which cause heavy strains so that they may be obviated or reduced to the lowest possible limit.

2 This limitation of the cutting capacity occurs in all metal cutting machines, although to a varying extent. While it is possible to increase the driving power of most machines ad libitum, and almost any amount of metal can be put into machine elements to give them rigidity, there are certain classes of machines where practical considerations limit such increase of power and strength. This is especially true in machines where the main elements have to be adjusted and handled with great frequency. The knee-and-column type of milling machine owes its success, to a large extent, to the ease and rapidity with which it can be manipulated and it is doubtful if it will ever be possible to increase the dimensions of the parts much beyond the present sizes, without losing the benefits of the peculiar construction of this type of machine. In order to increase the capacity of this type of milling machine, it becomes necessary to reduce the strains set up by the cut and there are only two elements which can be modified to accomplish this result. These are the hardness of the metal to be cut and the cutting qualities of the milling cutter. As it is impossible to control the first of these, the only avenue left for improvement leads in the direction of the milling cutter.

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

3 Experiments carried on at the works of the Cincinnati Milling Machine Company and extending over several years, starting with some isolated and almost desultory trials, and gradually becoming a series of carefully planned experiments, have led to results which are believed to be of general interest. These tests embraced spiral mills, end mills, both of the shell end-mill type and the spiral taper-shank type, side mills, slitting saws, face mills, and a new type of mill which, for lack of a better name, is called here a helical mill.

4 The action of the ordinary milling cutter is not a true cutting action, as it is commonly understood. By a true cutting action is meant, the driving of a wedge-shaped tool between the work and the chip and although this definition is not based on a generally accepted meaning of the term, it is believed that it expresses fairly well what most mechanics understand by cutting. Practically all milling cutters have their teeth radial and this, of course, excludes the possibility of driving a wedge between chip and work. The tooth compresses the metal until it produces a strain great enough to cause a plane of cleavage at some angle with the direction of the cutter. It then begins to compress a new piece, push it off, and so on. This at least *seems* to be the action of the cutter, judging by the form of the chips. These chips are in the form of needles or small bars.

5 The chip taken by a milling cutter varies very materially from that taken by a lathe or planer tool. These latter tools make chips of uniform section, whereas the section of a milling chip increases from zero to a maximum.

6 Fig. 1 shows a milling chip as it would appear, if no compression or distortion took place. The proportions are very much exaggerated, so as to bring its typical shape clearer into view. The width AB at the top is equal to the feed per tooth. The height BC is the depth of cut. The length BD is the width of cut. The section $MNOP$, shown half way on the chip, is a normal section and a measure of the amount of work which was done at the time the cutter passed the point M.

7 Fig. 2 shows the action of a milling cutter, with center O, when the cutter is rotating and the work is feeding at the same time. The tooth AB sweeps through the path BC. When the point B has reached the position B_1 a new tooth starts cutting. By this time O has advanced to position O_2, and the new tooth $A_2 B_2$ is not yet in a vertical position, when the point B_2 touches the work. When the cutter revolves, this point B_2 must penetrate into the work and compress the metal of the work. The result will be spring in the arbor. When this

spring has assumed certain proportions, the blade or tooth begins to remove a chip. This may be assumed to take place in the position B_3, the tooth simply gliding over the work from B_2 to B_3. This action must necessarily be very harmful to the cutter, and, it was believed that this, perhaps more than any other action of the cutter, caused its dulling. It would be especially severe with light cuts, as a relatively small amount of spring would allow the point B_2 to travel through a large arc. It would be quite possible that a tooth should fail entirely to take a chip, and that the succeeding tooth would take a chip of double the amount.

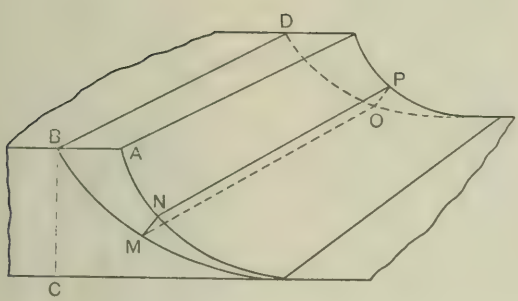


Fig. 1

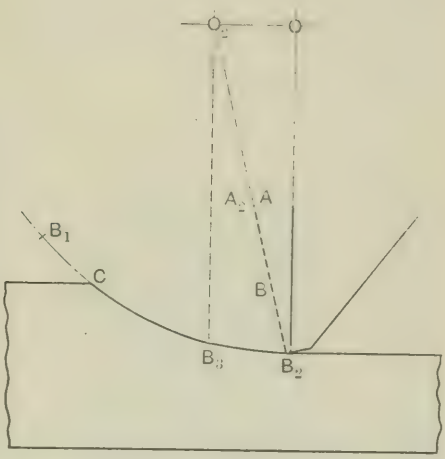


Fig. 2

Fig. 1 Metal Chip Assumed to be Produced by Milling Cutter without Distortion

Fig. 2 Diagram to Illustrate Action of Milling Cutter

8 This peculiar action of the milling cutter is inherent in its construction and cannot be avoided. The question then is how to minimize these harmful results.

9 Another feature, which limits the ability of a milling cutter to remove metal, is the proportion between the chip to be removed and the amount of space between two adjoining teeth. Such a limitation does not exist with lathe or planer tools, where the chips have unlimited space in which to flow off.

10 That this proportion between chip and chip space actually does form a limiting condition is well known and was brought most forcibly to the writer's attention when a large and powerful machine stalled, taking a cut in cast iron about $1\frac{1}{2}$ in. wide, $\frac{3}{16}$ in. deep and $12\frac{1}{2}$ in. feed per minute. Several times this amount of metal can be easily removed by the same machine, without sign of stress; yet the machine was incapable of removing more than 3 cu. in. of cast iron

per minute with this cut. Investigation showed that the amount of cast iron removed per tooth was sufficient to fill the chip space completely, and from that moment the action was like trying to push a solid bar of steel through a piece of cast iron. Another cutter, with more chip space, removed the same amount of metal with only a fraction of the power of the machine.

11 Similar instances occurred with gangs which had been in use a long time, and which had been ground down to such an extent that the chip space was materially reduced. This, combined with the fact that more highly developed milling machines led the shop to coarser feeds, showed that the ability of the machine to remove metal was not only governed by its power, but to an equal extent by the peculiarities of the milling cutter.

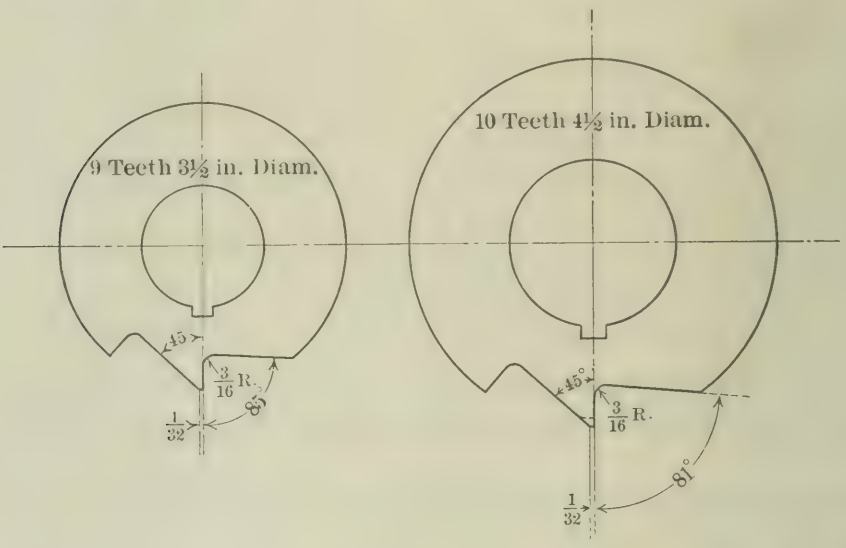


FIG. 3 FORM OF SPIRAL MILLING CUTTERS NOW USED BY THE CINCINNATI MILLING MACHINE COMPANY

12 The foregoing considerations led to a gradual evolution of spiral milling cutters. At first, the number of teeth of spiral mills was only slightly diminished, as it was thought that some element which was not considered might affect the result. Gradually the spacing was increased and the cutters, as now used, have taken the forms shown in Figs. 3 and 4.

13 Two standard sizes are used, although other sizes are required for special cutters and special gangs. The standard diameters are $3\frac{1}{2}$ in. and $4\frac{1}{2}$ in. The $3\frac{1}{2}$-in. diameter cutters are made with nine, and the $4\frac{1}{2}$-in. diameter cutters with ten teeth, which corresponds to a spacing of about $1\frac{1}{4}$ in. The point of the tooth has a land of $\frac{1}{32}$ in.,

and the back of the tooth forms an angle of 45 deg. with the radial line. The chip space is approximately four times as great as in the usual standard cutter of the present time and is formed with a $\frac{3}{16}$-in. radius at the bottom.

14 Though not directly connected with the foregoing, attention should be called to the fact that present practice calls for arbors which are too small. In the cutters shown here, the $3\frac{1}{2}$-in. cutter is made with $1\frac{1}{2}$-in. and $1\frac{3}{4}$-in. arbor, and the $4\frac{1}{2}$-in. cutter with $1\frac{3}{4}$-in. and 2-in. arbor.

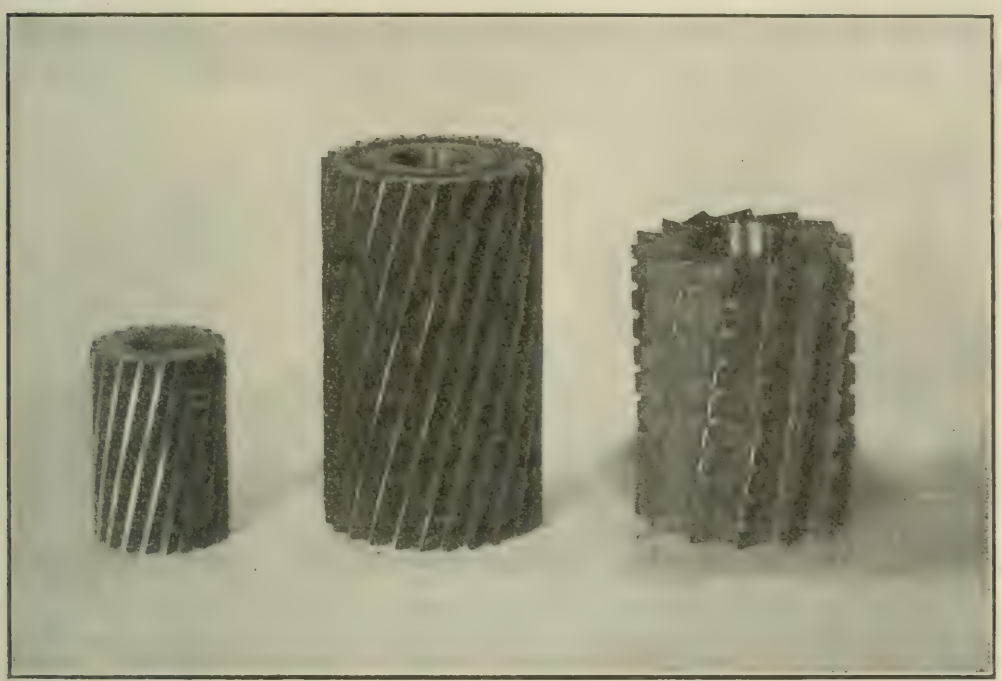
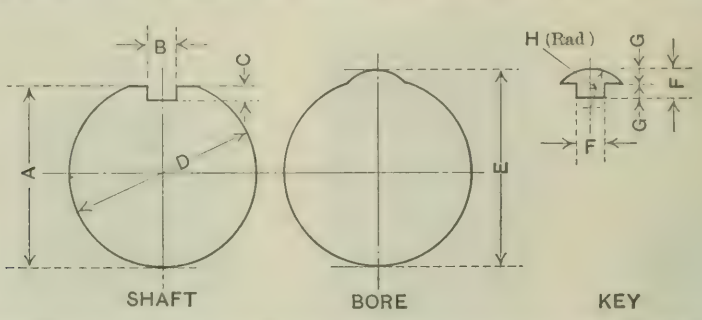


FIG. 4 COMPARISON OF OLD AND NEW STYLE SPIRAL MILLS

15 It is often very difficult to remove cutters from an arbor after they have done heavy work. It is frequently necessary in such cases to press the arbor out of the cutters. This sticking of the cutter is caused by the burring up of the key, and often the keyway in the arbor. For this reason, keys are used for gangs of cutters as shown in Fig. 5. A flat is milled on the arbor, and the keyway milled central with this flat. The flat portion of the key presses against the flat part of the arbor, and this effectively prevents burring. Cutters which are held on the arbor with such a key can always be very readily removed, even after prolonged and hard work. The keys are made out of a piece of round stock, grooved at both sides and then sawed apart.

16 Very satisfactory results were obtained with these cutters. Figs. 6, 7 and 8 show the results of tests made with cutters with $\frac{5}{8}$ in., $\frac{3}{4}$ in. and $1\frac{1}{8}$ in. spacing. Cuts were taken on cast-iron test blocks as shown in Fig. 9. The cross-sectioned part of the test block was milled out. A series of tests was made on the left-hand half of the block with one kind of cutter and on the right-hand half with another cutter. It will be noticed that the same amount of power is required to take a cut $\frac{1}{4}$ in. deep and with 10.4 feed with a cutter of $\frac{5}{8}$ in. pitch; and a cut $\frac{1}{4}$ in. deep and with 13.5 feed but with a cutter of $1\frac{1}{8}$ in. pitch.

17 It was not safe to assume that all test blocks would be of the same hardness. In order to correct whatever error there might be



A	*B*	*C*	*D*	*E*	*F*	*G*	*H*
0.973672	$\frac{5}{32}$	$\frac{5}{64}$	1	1.05179	$\frac{5}{32}$	$\frac{5}{64}$	$\frac{13}{64}$
1.03779	$\frac{5}{32}$	$\frac{5}{64}$	$1\frac{1}{16}$	1.11592	$\frac{5}{32}$	$\frac{5}{64}$	$\frac{13}{64}$
1.21875	$\frac{3}{16}$	$\frac{3}{32}$	$1\frac{1}{4}$	1.3125	$\frac{3}{16}$	$\frac{3}{32}$	$\frac{1}{4}$
1.4543	$\frac{1}{4}$	$\frac{1}{8}$	$1\frac{1}{2}$	1.5793	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{11}{64}$
1.7111	$\frac{1}{4}$	$\frac{1}{8}$	$1\frac{3}{4}$	1.8361	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{11}{64}$
1.94734	$\frac{5}{16}$	$\frac{5}{32}$	2	2.10359	$\frac{5}{16}$	$\frac{5}{32}$	$\frac{13}{32}$

FIG. 5 SHAPE AND DIMENSIONS OF KEYS USED FOR MILLING-CUTTER ARBORS

on account of unequal hardness of the test blocks, hardness tests were made of the different blocks. These consisted in taking a cut $\frac{3}{16}$ in. deep and with various feeds on each one of the blocks, and finally a check test on the first block, to make sure that the cutter had not appreciably dulled.

18 It will be seen from these diagrams that there is a large increase in the amount of metal which can be removed with the same amount of horsepower, by using these wide-spaced cutters; and that, therefore, the scope of the knee-and-column type of milling machine has been enlarged without increasing sizes or weight of parts and thus decreasing the handiness of the machines.

19 Though increased capacity for removing metal is one of the main advantages of this form of cutter, there are others of consider-

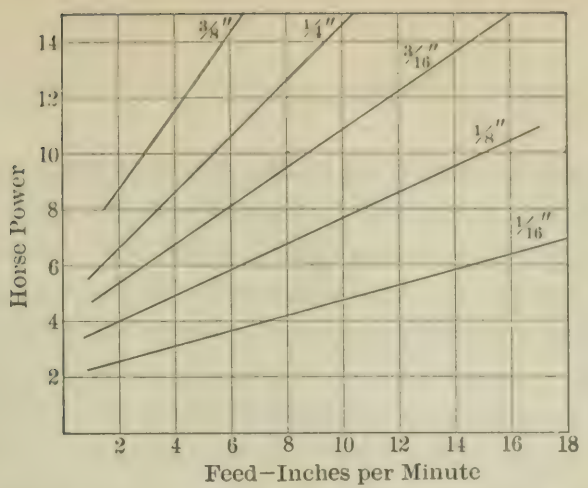


FIG. 6 TESTS OF SPIRAL NICKED MILLING CUTTER $3\frac{23}{32}$ IN. IN DIAMETER; 18 TEETH AND ABOUT $\frac{5}{8}$ IN. PITCH. CUTTING CAST IRON CORRECTED FOR HARDNESS. WIDTH OF CUT 6 IN.

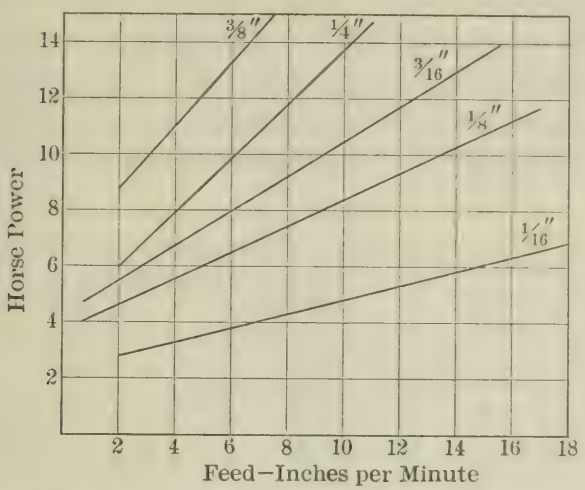


FIG. 7 TESTS OF SPIRAL NICKED MILLING CUTTER $3\frac{1}{12}$ IN. IN DIAMETER; 14 TEETH AND ABOUT $\frac{3}{4}$ IN. PITCH. CUTTING CAST IRON, CORRECTED FOR HARDNESS. WIDTH OF CUT 6 IN.

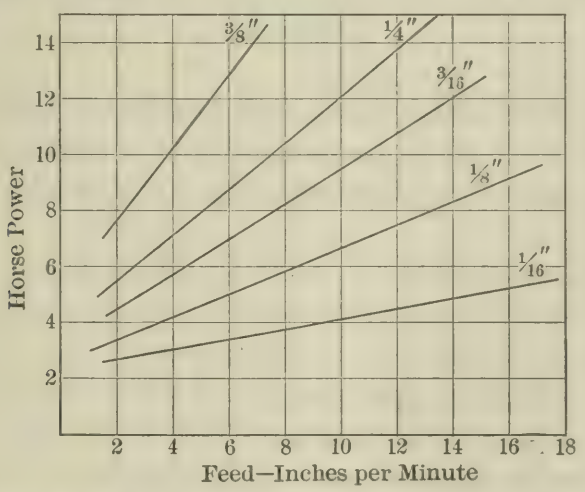


FIG. 8 TESTS OF SPIRAL NICKED MILLING CUTTER 6 IN. IN DIAMETER; 16 TEETH AND ABOUT $1\frac{1}{8}$ IN. PITCH. CUTTING CAST IRON, CORRECTED FOR HARDNESS. WIDTH OF CUT 6 IN.

able importance. It was found that for roughing on the ordinary work in the shop a cutter with the wider-spaced teeth would remain sharp for a longer period, notwithstanding that feeds had been increased. The system of the Cincinnati Milling Machine Company requires all gangs and cutters to be re-sharpened after a lot of pieces has been milled. It used to be necessary, at least on the larger lots, to re-sharpen the gang once and sometimes twice for one lot, or, if this was not deemed advisable, the feed had to be reduced for at least part of the pieces, in order to make the cutter last during the entire lot. In all cases where the wide-spaced cutters were used, the entire lot was run through without re-sharpening the cutter or reducing the feed; and it should be kept in mind that this feed was from 25 to 100 per cent greater than previously used. There is no case on record where the cutter or gang was dull at the end of the lot, so that our observations as to the endurance of the cutters are incomplete. However, it is perfectly safe to say, that in all cases under observation the cutter

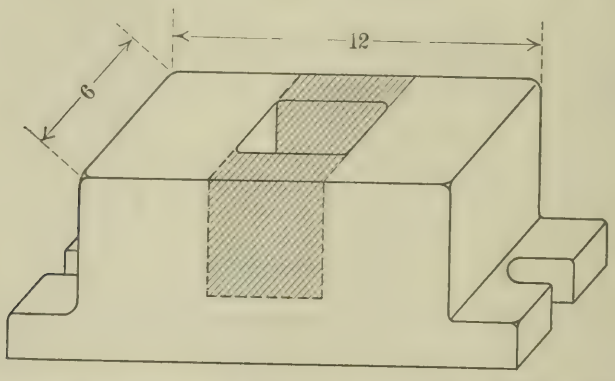


FIG. 9 TEST BLOCK USED IN TESTING CUTTERS WITH DIFFERENT SPACING OF TEETH

maintained its sharpness longer; that in a great many cases double the amount of work could be done without re-sharpening; and that there is good reason to believe an even greater gain than this was obtained.

20 A further advantage is, that as these cutters have approximately only half the number of teeth of what is now considered a standard cutter, the time for re-sharpening is only half as much.

21 It was pointed out that the ratio of pitch to depth is practically the same as in the present standard cutter, so that the depth of tooth is practically doubled and this cutter can be sharpened much more frequently than the present standard cutter. Consequently the life of the cutter has been much increased, probably more than doubled.

22 A glance at the drawing of these cutters gives the impression that the teeth are weak and the writer has watched this feature with great care. The cutters themselves, however, do not give this impression; on the contrary, they look stout and well proportioned. They have been subjected to the heaviest class of work and many times were purposely abused in order to find their weak points; yet there is no case on record that any of them have broken although they have been used for more than two years and all breakages of cutters are carefully noted. On the other hand, breakages of the old cutters are not at all infrequent.

23 Though these cutters are capable of removing metal more rapidly than the older type of cutter there are many cases where this feature cannot be taken advantage of, as, for instance, where light work is to be done or a small amount of stock to be removed. In such cases, however, the metal is removed with less power and consequently with less strain on the machine and the life of the machine is lengthened without limiting its output.

24 With the wide spacing of the teeth it may seem that there would be cause for apprehension as to the action of the feed. It seems as if the feed would be liable to act with jerks. This, however, is not the case. On the contrary, the feed is smoother and there is less of a jerk when the cutter first strikes the work, probably because there is less spring in the arbor and less tendency for the cutter to ride over the work, as will be explained later in connection with the description of cutters.

25 In connection with this, it is interesting to note that when cast iron is milled by these wide-spaced cutters, it appears to be very soft and when the same piece is milled by an old-style cutter, it appears to be much harder. When using the wide-spaced cutter, there is a notable absence of jerking, chattering and of the peculiar singing noise which is so often noticed on milling machines.

26 There is, of course, a difference in the hardness of different pieces of cast iron, and many recommendations as to the proper feeds and speeds for milling cast-iron work, made by the writer for his company, were looked askance at. The impression seemed to prevail that feeds and speeds which were possible on American iron, were out of the question on European iron (especially English and German); and again, that feeds and speeds proper in America for western iron were not suitable for eastern iron. To test the truth of the matter, a number of bars of cast iron were obtained from different foundries in America, England, France and Germany.

These bars covered a great many mixtures and makes, and the difference between English and American, or German and American iron, or between eastern and western iron in America, was found to be no greater than that between different specimens of western American iron. Even German Spiegeleisen, famous for its hardness, cut just as freely as soft western iron, and required but little more power. However, it did require more clearance, wide spaces, and a low speed.

27 These wide-spaced cutters were originally intended for roughing operations only, but the very satisfactory finish obtained when roughing led to the use of the cutters for finishing also. If there is any difference at all in the finish produced, the advantage is on the side of the wide-spaced cutter. The fact that this wide-spaced cutter will cut a greater number of pieces without dulling, means, of course, that the average finish of an entire lot is better.

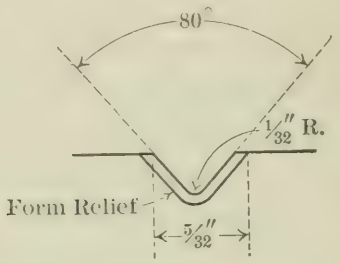


FIG. 10 CHIP BREAKER, DOUBLE SIZE

28 It is generally believed that for finishing alone a milling cutter should be used without chip breakers, the effect of the chip breaker being to scratch the surface. To overcome this trouble, chip breakers are made as shown in Fig. 10 with clearance at both corners. This prevents the tearing up of metal with the result that a cutter with these chip breakers produces as good a finish as one without chip breakers.

29 It should be pointed out that this form of chip breaker has an advantage also for roughing cuts. The point of the cutter, where the unrelieved side of the chip breaker drags over the work, is the first point to give out. Making the chip breaker with clearance on both edges prolongs, therefore, the life of the cutter.

30 One of the great advantages of this form of chip breaker is, that one gang can be used for both roughing and finishing. A great many, if not most milling operations, call for two chuckings, one for roughing, and one for finishing. This will be found to be necessary

wherever much metal is to be removed, on account of distortion caused by the cut, the heavy clamping required, heating, spring of arbor or fixture and the unbalanced condition of the work after the scale has been removed on one side. In order to do the roughing as rapidly as possible chip breakers are required; and in order to get proper finish, it has heretofore been necessary that the finishing gang be without chip breakers. It paid, therefore, to have two gangs whenever the number of pieces to be milled was sufficiently large, but this involved considerable extra expense for cutters. The new form of chip breaker, however, permits using one gang for both finishing and roughing.

31 It is a common belief that better finish can be obtained with teeth closely spaced, but experience with the wide-spaced cutter shows that there is no ground for this belief. The grade of finish may be expressed by the distance between successive marks on the work. These marks are revolution marks and not tooth marks. It is practically impossible to avoid these revolution marks. They are caused by the cutter not being exactly round or quite concentric with the hole, by the hole not being of exactly the same size as the arbor, by the arbor not being round, by the straight part of the arbor not being concentric with the taper shank, by the taper shank not being round or of the same taper exactly as the taper hole in the spindle, by this taper hole being out of line with the spindle, by looseness between the spindle and its bearings, etc. Each of these items is very small in any good milling machine; yet the accumulation of these little errors is sufficient to cause a mark and this mark needs to have a depth of only a fraction of a thousandth of an inch to be very plainly visible. As these marks are caused by conditions which return once for every revolution of the cutter, it is plain that the spacing of the teeth can have no effect on the distance between them and, therefore, on the grade of finish.

32 To test this still further, two cutters of exactly the same size were placed side by side on an arbor. The cutters were ground together so as to be sure they were of equal diameter and they were ground on the arbor so as to be sure that the error would appear simultaneously for both cutters. A block of cast iron was finish-milled with these cutters in such a way that each cutter would sweep half the width of the block. The same number of marks appeared on both sides of the block, and these marks were exactly in line with each other, as might have been expected. The grade of finish was the same for both sides. It was neglected to mark the two sides of

the casting to show which cutter was operating. After this test, all but one of the teeth of one of the two cutters were ground lower, so as to be out of action entirely, leaving only one tooth of one cutter operative. Another cut like the first one was taken over the same block, and again the finish appeared the same on both sides. There

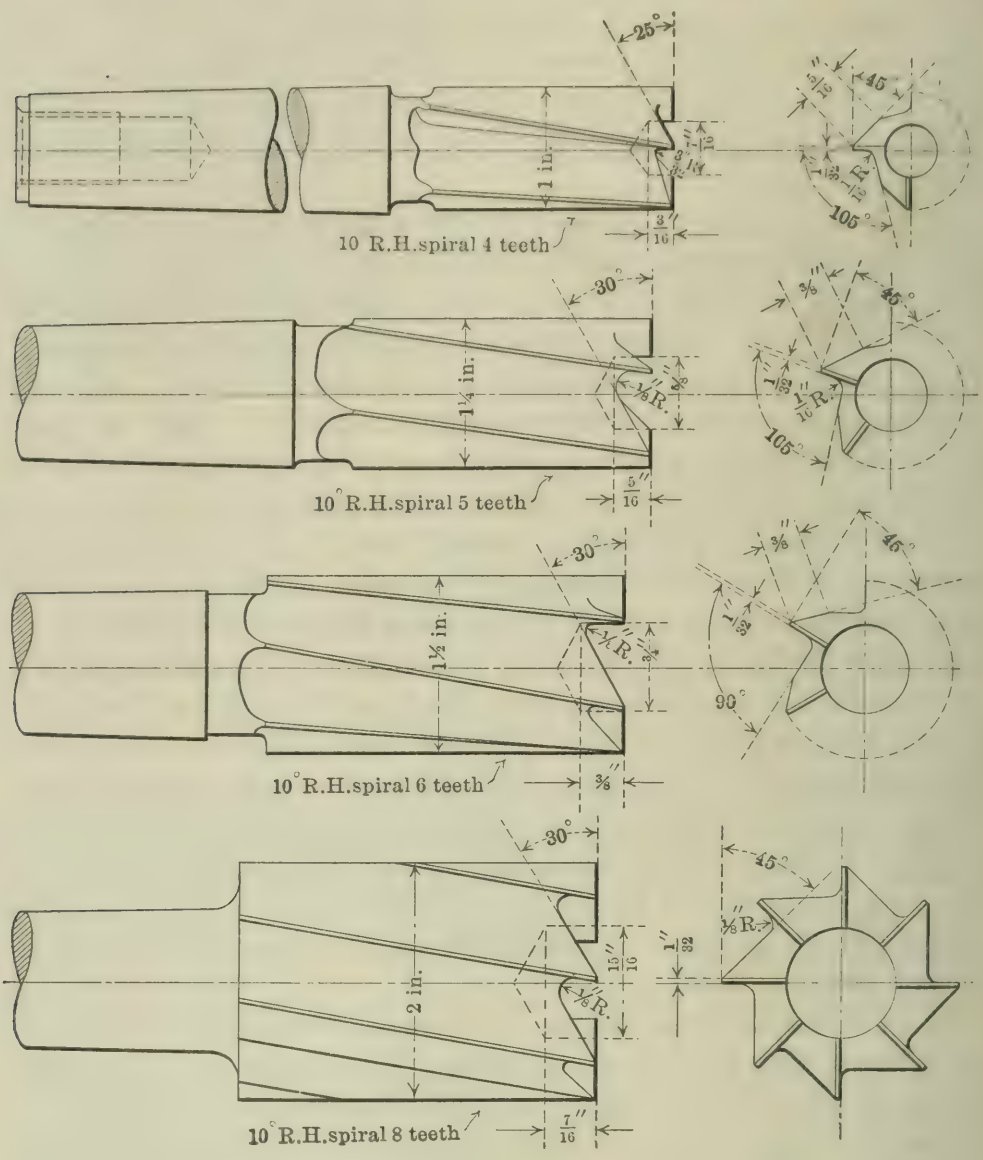


FIG. 11 NEW TYPE OF TAPER SHANK END MILLS

was a difference of opinion between different observers as to which side was cut by the single tooth. By close observation, however, a difference could be detected when light fell on the work in a certain direction, under which conditions one side showed more gloss than

the other. Straightness, flatness and smoothness to the touch were exactly the same for both sides, notwithstanding that one cutter had one tooth only and the other fourteen teeth. Though it is not recommended here to use cutters with one tooth only for finishing, the foregoing test shows plainly that there is no merit in fine spacing. Attention is again called to the fact, that even though the finish on a single piece *might* be better with more teeth in action, the *average* finish for an entire lot of pieces is better with less teeth.

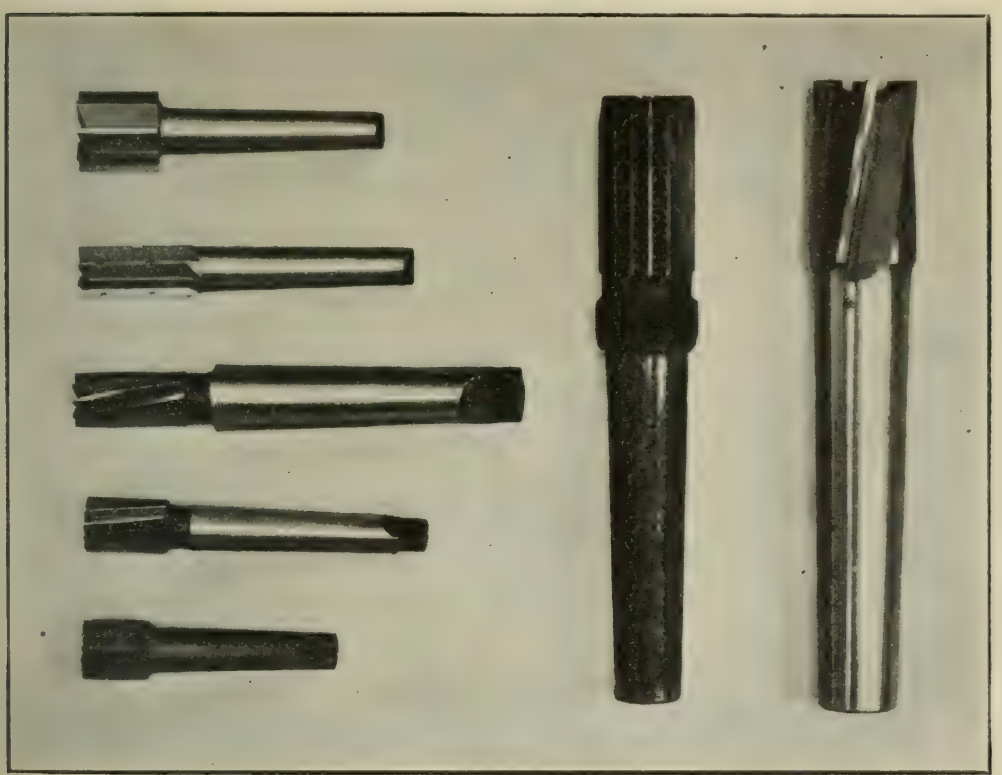


FIG. 12 COMPARISON OF OLD AND NEW STYLE END MILLS

33 Figs. 11 and 12 show the end mills which are now considered standard by the Cincinnati Milling Machine Company and which fill practically all requirements. They are made in sizes of 1 in., 1 $\frac{1}{4}$ in., 1$\frac{1}{2}$ in. and 2 in. in diameter, the smallest with four, and the largest with eight teeth. It will be noticed that in order to preserve the strength of the teeth it is necessary to mill the back of the teeth of the three smaller sizes with two faces. A number of tests have been made with these cutters, but no comparative tests as to power consumption. Their action is remarkably free. This was clearly demonstrated by the following experiment: A 2-in. taper shank end-

mill milled a slot $1\frac{1}{16}$ in. deep in a solid block of cast iron at a rate of 6 in. per min. The block was clamped to the table of the milling machine and the knee was fed upward. Under these conditions the

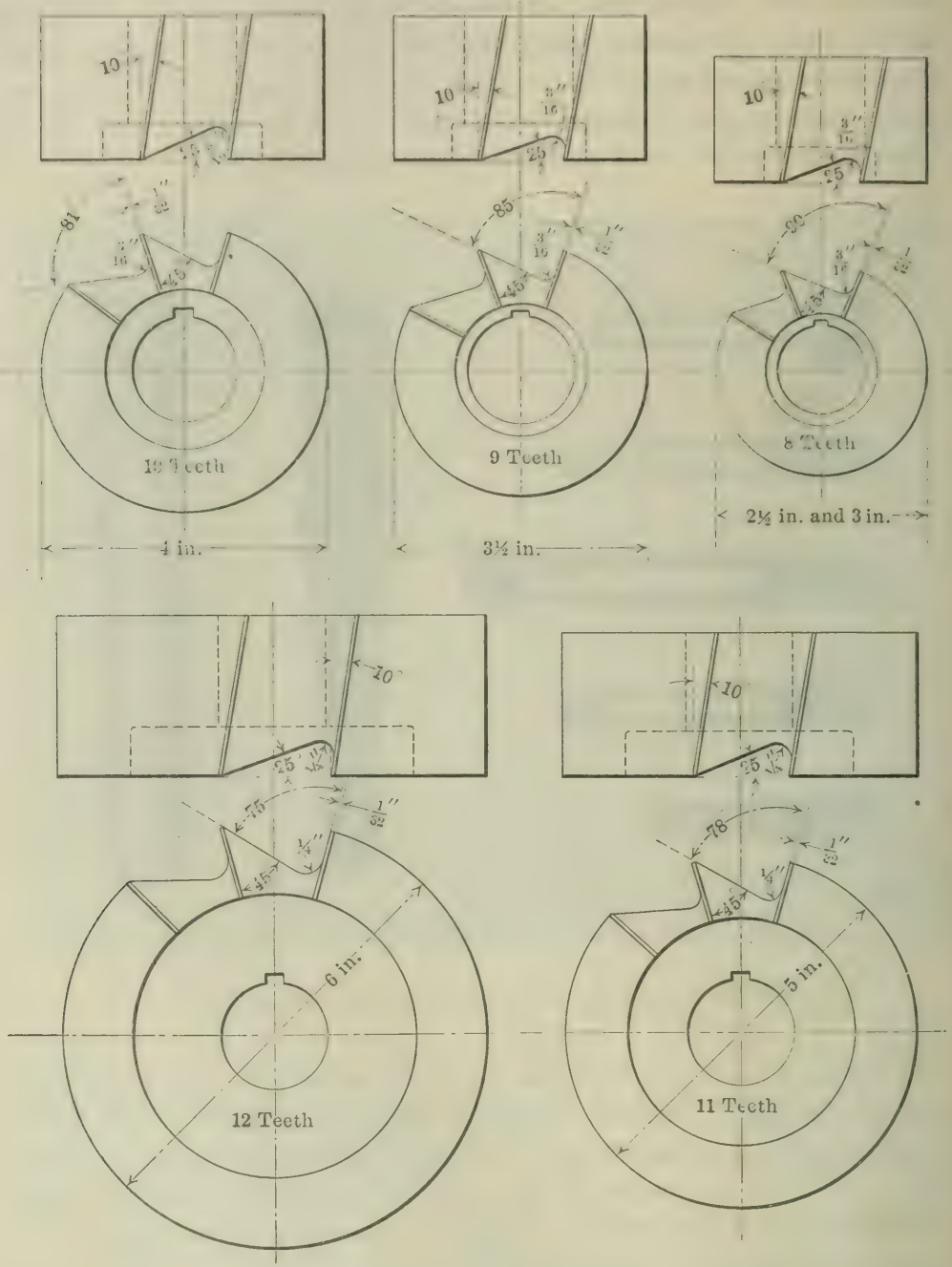


FIG. 13 NEW TYPE OF SPIRAL SHELL CUTTERS

chips did not free themselves from the cutter, but were carried around and ground up. The cutter was cutting over half its circumference. These two conditions combined make the task for the milling cutter

about as difficult as is imaginable. There was, however, no sign of choking and the power consumption was not higher than it would have been with a spiral mill under ordinary conditions. The same cutter would remove from the end of the casting a section $1\frac{1}{2}$ in. wide

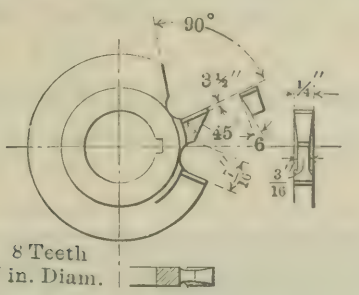


FIG. 14a DETAILS OF NEW TYPE OF SIDE MILLS

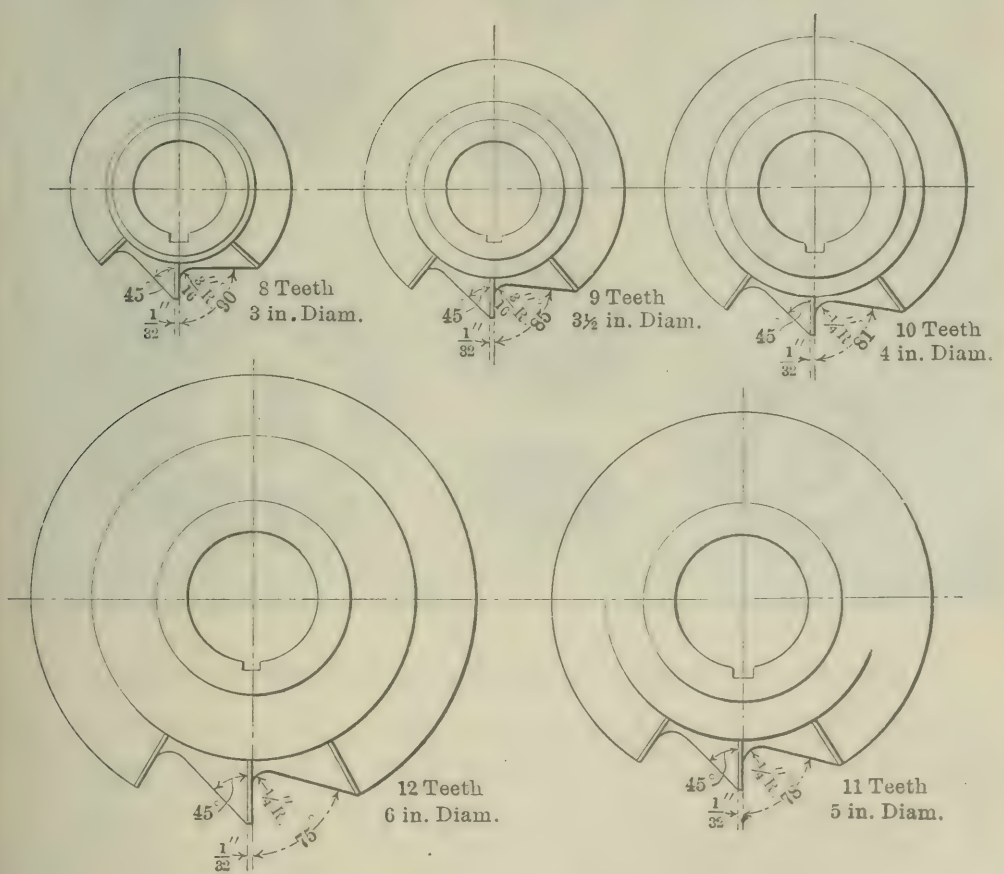


FIG. 14b NEW TYPE OF SIDE MILLS

and $1\frac{1}{2}$ in. deep. Under those conditions, the chips would free themselves from the cutter and these chips were rolled up in pieces much like the chips obtained from a broad planer tool, when taking a finish-

ing cut. This cut was taken with a feed of 11 in. per minute. Another similar cut, but 1 in. and $1\frac{1}{8}$ in. in section was taken with a feed of 33 in. per minute. Similar though much lighter cuts were taken with ordinary end mills, and in the same piece of cast iron. Again the cast iron seemed to be very hard, and became glossy when cut with an ordinary cutter, but appeared to be soft when cut with the wide-spaced cutter.

34 Fig. 13 shows the shell end mills of the wide-spaced type, which are now considered standard for their use by the Cincinnati Milling Machine Company.

FIG. 15 COMPARISON OF SIDE MILLS AND SLOTTING MILLS

35 Figs. 14*a* and 14*b* show the side mills and Fig. 15 gives a comparison of the new and old style side mills and slotting mills.

36 Face mills have also undergone a gradual evolution and they are now used by the company and catalogued, though not made by them for use of customers, as shown in Fig. 16. Fig. 17 shows a cutter of a design now generally considered to be standard. In this latter design, the blades are spaced 1 in. apart, or approximately so; they are set radial, and have no means to keep them from pushing back except the regular holding means. The wide-spaced face mill, on the other

hand, has the blades spaced 2 in. apart. They are set at an angle of 15 deg. with the radial line, and are backed by a backing ring with a set screw for each blade. These set screws allow the blade to be adjusted, besides forming a stop against upward movement under pressure. A face mill may be considered as a planing tool moving in a circular path. The cutting edge, therefore, is axial and not radial. To set the blades at an angle with the axis does not produce rake. The wide-spaced face mill shown here has rake, because the blades are set at an angle with the radial line.

37 It will be noticed that the blades are set at an angle with the axis. It will further be noticed, in the enlarged view of the blades, showing the rounded corners, that the corners are not provided with a round, but rather with three faces, which together approximate a curve. It is to offset the effect of this round that the angle with the axis is introduced. In Fig. 18 a new type of face mill is shown at the left and at the right a mill of the old or regular type.

38 However accurately a milling machine may be built, the spindle is not exactly at right angles with the table. The amount of variation from the right angle is very small in a properly built machine, but some variation exists. Besides, this variation is liable to become greater when the machine wears. The result is, that when feeding in one direction the leading teeth of the cutter dig deeper into the work, leaving the other side of the cutter entirely clear, but when feeding in the opposite direction the opposite takes place, which makes the teeth drag over the work. In order to provide the teeth with clearance, the back end of the tooth is ground away at an angle of three to five degrees.

39 It will further be noticed, that there is a land of $\frac{3}{16}$ in. only where the blade is straight. It is the excess of width of the cutting blades which is liable to cause chatter. Strange as it may seem, this chatter is more pronounced with a light than with a heavy cut. It is not meant that there is actually chatter, but merely that when there is a tendency to chatter, the tendency is greater on a lighter cut. The cause is that the tooth does not enter the work but tries to ride over it. When the cutter has been lifted sufficiently, the pressure becomes great enough to make the blades enter. The next blade meets the same difficulty about entering, is lifted again, and so on. This action causes a series of radial chatter marks and is very much worse with wide blades than with narrow ones; and again very much worse with a large number of blades than with a few. A $\frac{3}{16}$-in. land proved to be an acceptable compromise, as a wider land would quickly

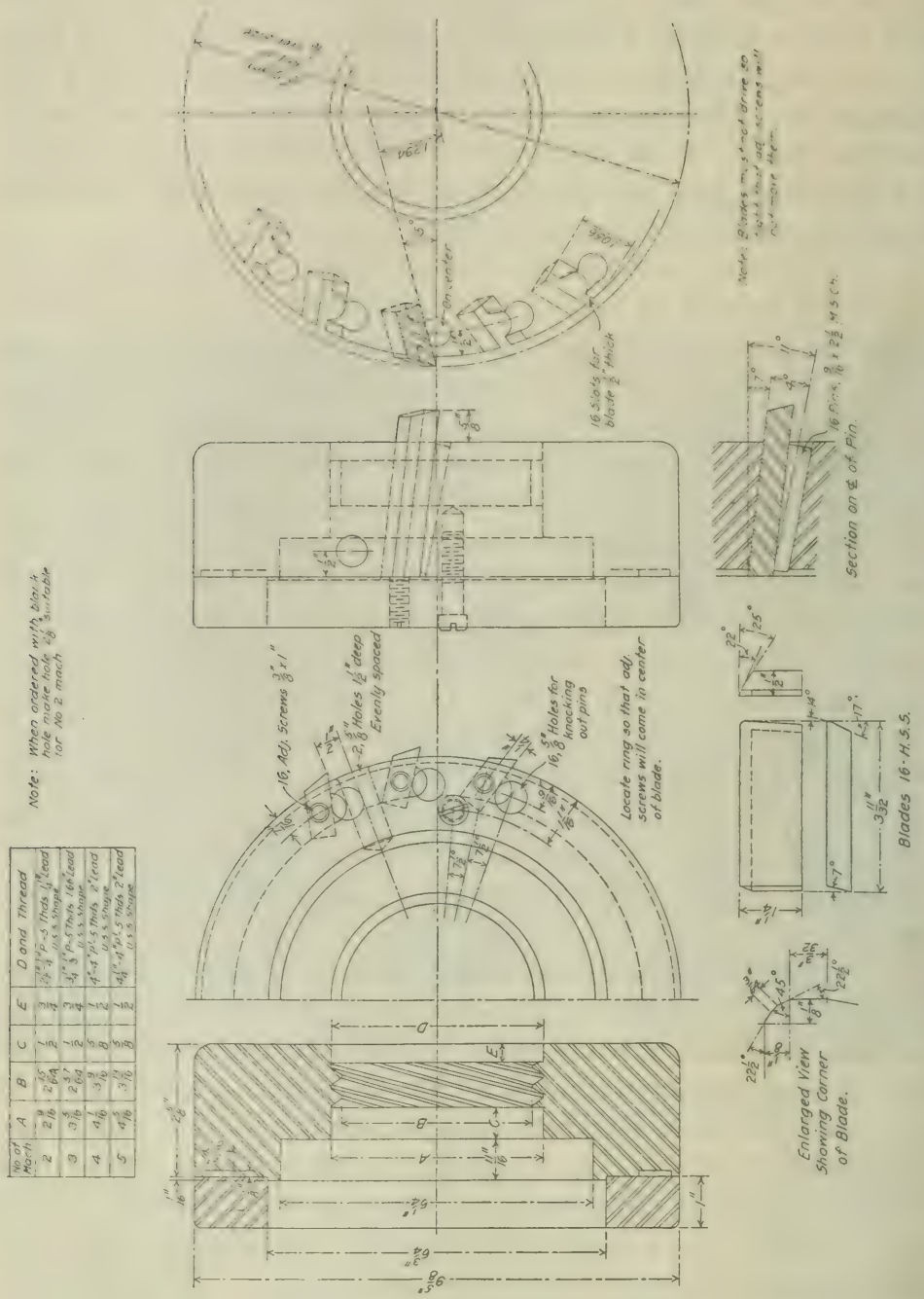


FIG. 16 TEN-INCH BLADE FACE MILL FOR HIGH-POWER MACHINES

dull the cutter, even if it did not make a chatter mark, while a narrower land would have the tendency to produce a scratchy finish.

40 In Fig. 19 is shown details of a helical cutter. These cutters consist of a cylindrical body, with two or three screw threads wound around them, the threads being of a section clearly indicated in the engraving. The helix is wound around the body with an angle of 69 deg. with the axis. The diameter is $3\frac{1}{2}$ in. and the lead of the helix $4\frac{1}{4}$ in. They are made in two styles, either single, or as interlocking right and left hand cutters. They are made with a rake of 15 deg.

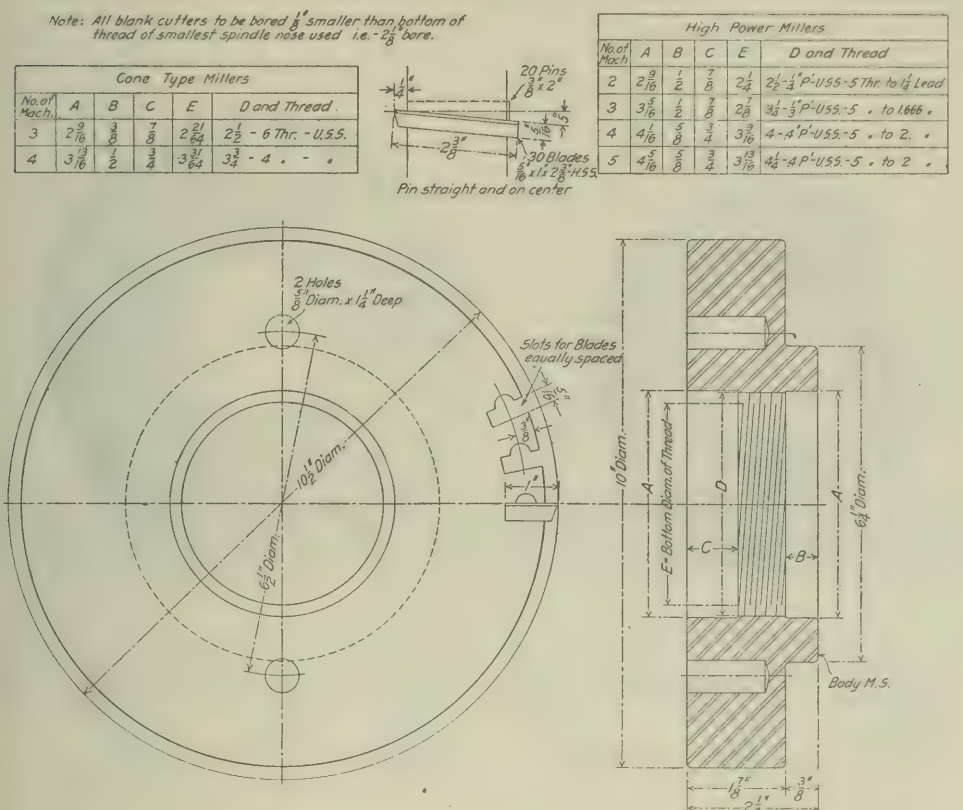

Note: All blank cutters to be bored 1/8" smaller than bottom of thread of smallest spindle nose used i.e.- 2 1/4" bore.
Cone Type Millers
No. of Mach. | A | B | C | E | D and Thread
3 | 2 9/16 | 3/8 | 7/8 | 2 21/64 | 2 1/2 - 6 Thr. - U.S.S.
4 | 3 13/16 | 1/2 | 3/4 | 3 31/64 | 3 3/4 - 4 " "
High Power Millers
No. of Mach | A | B | C | E | D and Thread
2 | 2 9/16 | 1/2 | 7/8 | 2 1/4 | 2 1/4 - 4 P - U.S.S. - 5 Thr. to 1/4 Lead
3 | 3 5/16 | 1/2 | 7/8 | 2 7/8 | 3 1/2 - 3 P - U.S.S. - 5 " to 1.666 "
4 | 4 1/16 | 5/8 | 3/4 | 3 9/16 | 4 - 4 P - U.S.S. - 5 " to 2. "
5 | 4 5/16 | 5/8 | 3/4 | 3 13/16 | 4 1/4 - 4 P - U.S.S. - 5 " to 2 "
20 Pins 3/8" x 2"
30 Blades 5/16" x 1/2" x 2 3/8" H.S.S.
Pin straight and on center
2 Holes 5/8" Diam. x 1 1/4" Deep
Slots for Blades equally spaced
10" Diam.
10 1/2" Diam.
6 1/2" Diam.
6 1/4" Diam.
Body M.S.

FIG. 17 FACE MILL OF OLDER TYPE

and clearance of 5 deg. when used for steel, and with a rake of 8 deg. and clearance of 7 deg. when used for cast iron. Their most distinguishing feature is, that they push the chip off in the direction of the axis of the cutter, or at right angles to the feed. The power consumption is extremely low for steel, but does not show up so favorably for cast iron. A roughing cut in steel requires only about one-third the power of an old-style spiral mill. Another distinguishing feature is, that this cutter does not make revolution marks but tooth marks. As

a result, a much coarser feed can be used for finishing. A cutter with three teeth will allow of a finish three times as fast as an ordinary spiral mill. Still another feature of this cutter is the entire absence of spring in the arbor when cutting steel. It is possible to take a finishing cut over a piece of steel, then return the work under the cutter and let the cutter revolve any length of time without producing a mark. Fig. 20 shows how this feature was made use of in the milling of steel test pieces.

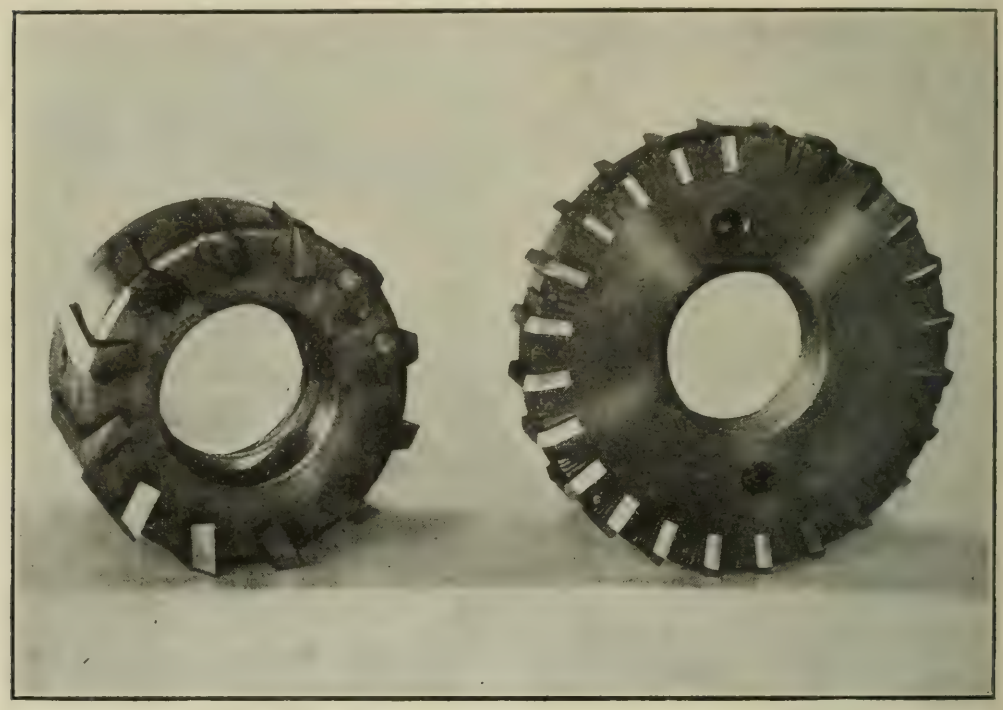


FIG. 18 COMPARISON OF HIGH POWER AND REGULAR FACE MILLS

41 It was originally thought that a single cutter of this description would do well for finishing, but not for roughing, on account of the excessive end pressure on the spindle, and the interlocking cutter was made to obviate this end pressure. However, it was found that this end pressure, though perceptible, was no disturbing element. Cuts which required 80 amperes with the interlocking cutter, required 85 amperes with the single cutter. In order to see if continued use of the single cutter would cause increasing friction at the spindle end, a great number of cuts were taken in as rapid succession as it was possible to adjust the machine for the next cut.

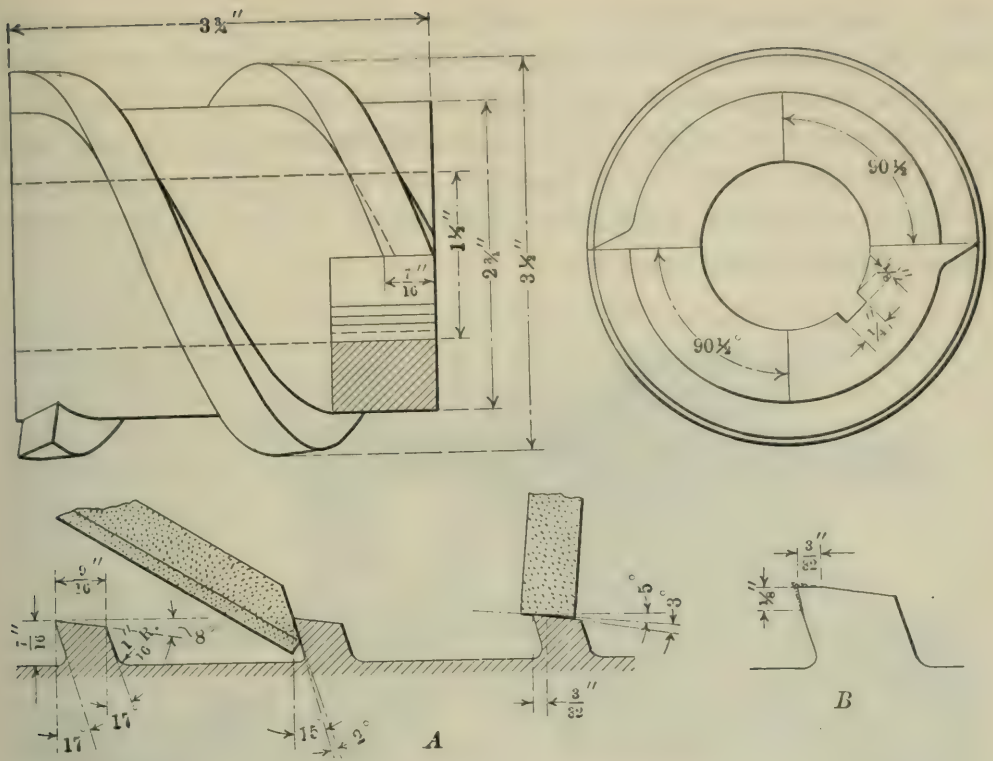


FIG. 19 DETAILS OF NEW TYPE OF HELICAL CUTTER

42 The fact that there is no spring in the arbor makes it possible to use the milling machine without braces in a great many cases where they would otherwise be needed.

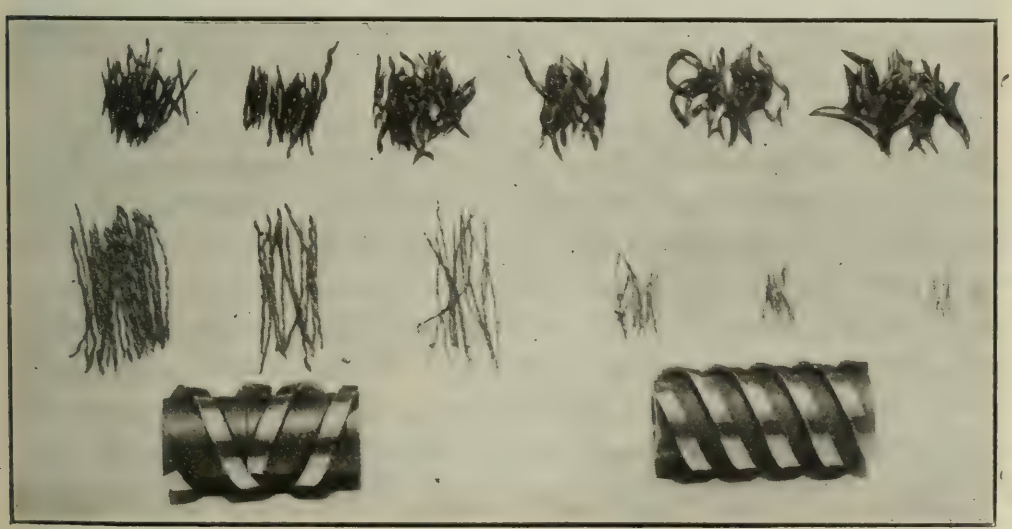


FIG. 20 HELICAL CUTTERS, SINGLE AND INTERLOCKED, AND CHIPS PRODUCED BY THEM

43 The chips come from the work in the form of gimlets as shown in Fig. 20. The back of the chip is polished or burnished, and a surface of the work shows no sign of tearing of the metal.

44 It was first believed that these cutters would work best at a high speed; but it was found that this was not the case. They produce the best results when run at the same number of revolutions as the ordinary spiral mill.

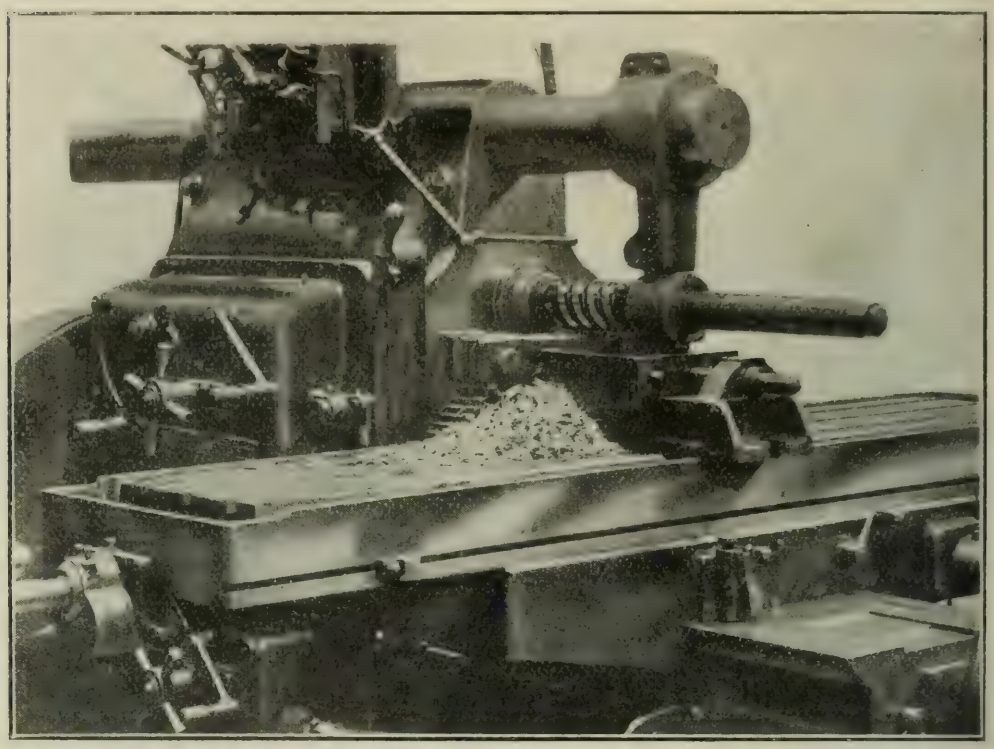


FIG. 21 HELICAL CUTTER AT WORK ON STEEL TEST PIECES

45 The writer believes that the remarkably low-power consumption is due to what might be called "virtual rake," which is an angle depending on the angle of rake, and on the angle the thread or tooth makes with the axis. This virtual rake becomes a small angle when the actual rake is small. This is the case with the cutter as used for steel, where the actual rake is 75 deg. Where, however, the angle of rake approaches 90 deg., the influence of the helix becomes very much less pronounced; and, if the actual rake were 90 deg., the influence of the spirality would be zero; in other words, the virtual would equal the actual rake. This may explain why the saving in power consumption is not so pronounced when cutting cast iron. It is believed that this saving of power would be equally as great with cast

iron as with steel, if the same virtual rake could be obtained, and this supposition was borne out by a few tests made on cast iron with a helical cutter ground for steel. The fact, however, that the edge of the cutter would not stand up, made it impossible to extend the tests far enough to come to a safe conclusion.

46 Another reason which suggests itself to the writer, as to why the helical cutter shows less saving in power on cast iron than on steel, is the result of a series of tests made on cast iron and steel with spiral mills with and without rake, the rake being in all cases 9 deg. These cutters showed improved efficiency for steel and cast iron, but much more for the first than for the latter. A cutting tool must detach the chip from the work, bend the chip and at least partially break it up. When cutting steel, the radius of curvature of the chip becomes greater with increased rake and the extent to which the chip is broken up becomes less. Cast iron will stand much less bending before breaking, so that, even with increased rake, the chip is still broken up as before, and no saving in power can be effected in this part of the process.

47 Figs. 22 and 23 are diagrams comparing the performance of different styles of cutters for different materials and the different depths of cut. Fig. 22 gives a comparison with feeds of 4 in. per minute and Fig. 23 for 14 in. per minute. It will be noticed that all lines are practically straight with the exception of the line for the regular face mill when cutting machinery steel. This line makes a sharp turn. This is believed to be due to the fact that the blades of this face mill did not project far enough beyond the body. As cast iron chips were crumbled up the effect was not noticeable for cast iron, but became quite important for machine steel. Fig. 22 shows that for cutting cast iron the high-power face mill is the most efficient. Then comes the regular face mill, then the spiral mill with $1\frac{1}{8}$-in. spacing, then the spiral mill with $\frac{3}{4}$-in., then the spiral mill with $\frac{5}{8}$-in. spacing. The 5-in. and 3-in. end mills come last in efficiency. These mills are of the old type with relatively fine spacing.

48 The order of efficiency of the different cutters is somewhat different for machine steel. The helical mill comes first, then the high-power face mill, then the spiral mill with $\frac{3}{4}$-in. spacing (no tests were made with spiral mills with $\frac{5}{8}$-in. and $1\frac{1}{8}$-in. spacing on machine steel) and finally the regular face mill; but it should be noticed that, if the curve for this mill had continued the way it started, it would have been below the curve for a spiral mill.

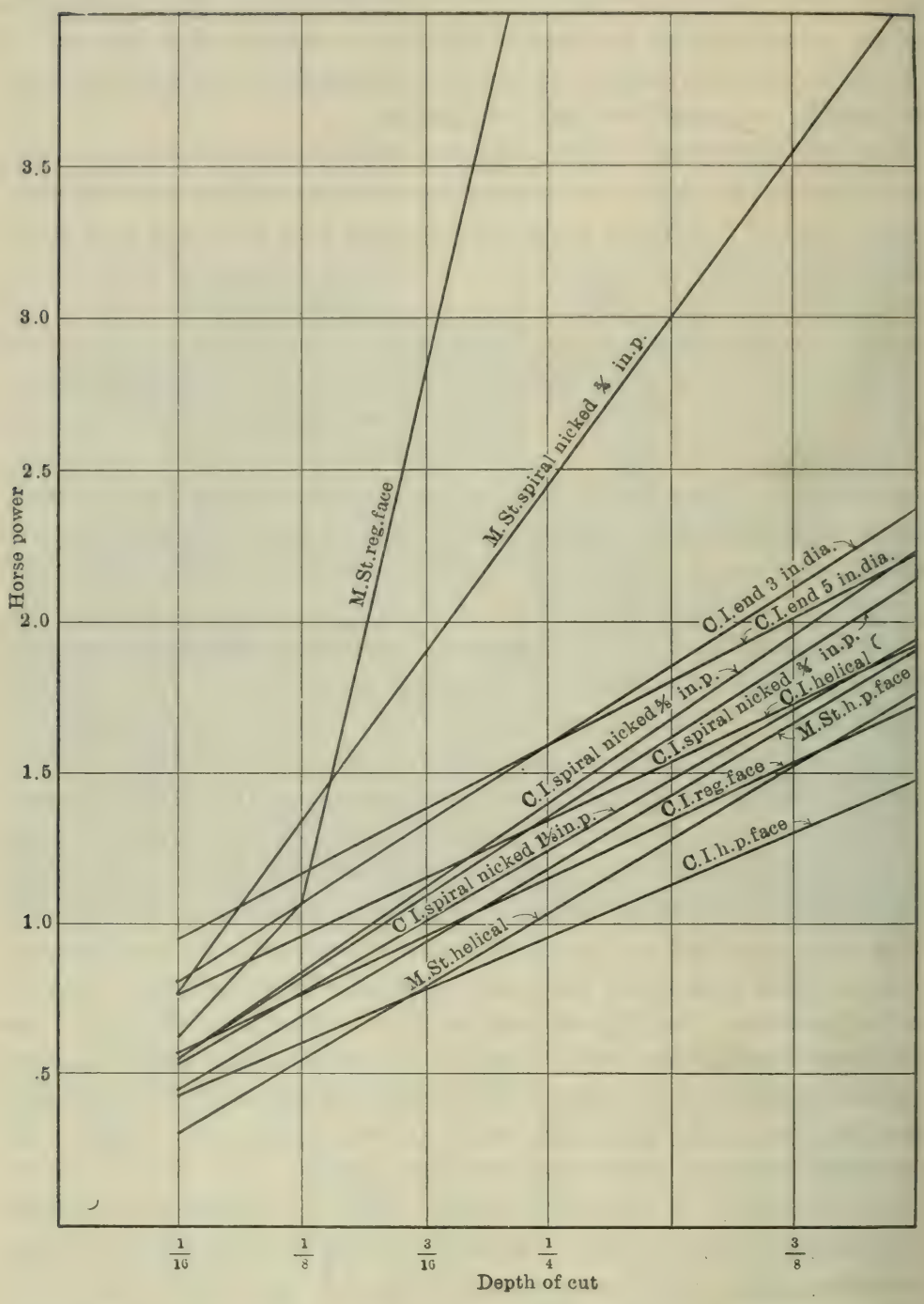


FIG. 22 EFFICIENCY TESTS OF CUTTERS. COMPARISONS CURVE REDUCED TO 1 IN. WIDTH OF CUT. CAST-IRON CURVES CORRECTED FOR HARDNESS OF MATERIAL. FEED 4 IN. PER MIN.

49 Fig. 23 gives comparative curves for a feed of 14 in. per min. The order of efficiency is much the same as in Fig. 22 with some ex-

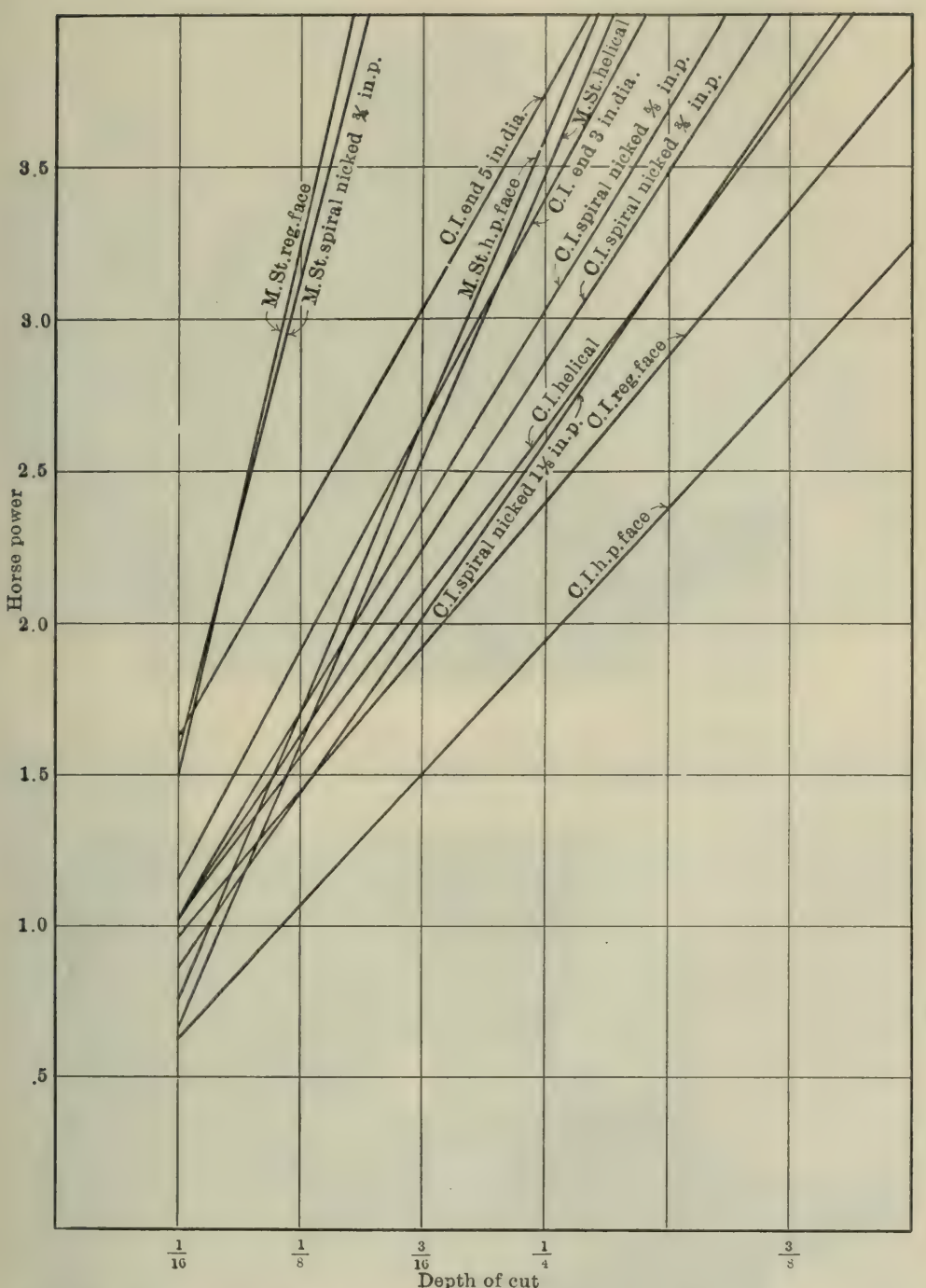


FIG. 23 EFFICIENCY TESTS OF CUTTERS. COMPARISON CURVES REDUCED TO 1 IN. WIDTH OF CUT. CAST-IRON CURVES CORRECTED FOR HARDNESS OF MATERIAL. FEED 14 IN. PER MIN.

ceptions. The helical mill, for instance, becomes more and more efficient as the heavier cuts are taken.

APPENDIX

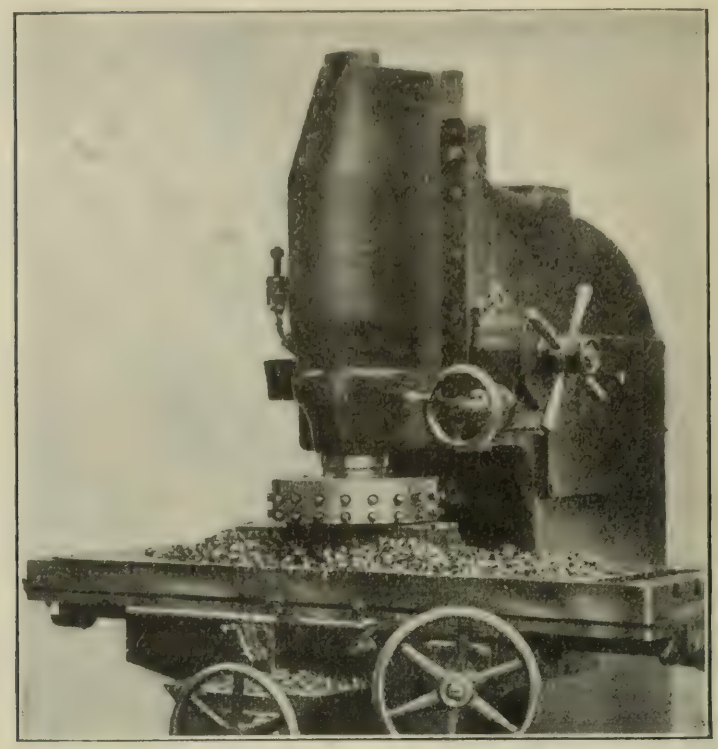


FIG. 24 HIGH-POWER FACE MILL OF EARLY CONSTRUCTION, MILLING MACHINE-STEEL TEST BLOCKS. CHIPS RESEMBLE PLANER CHIPS.

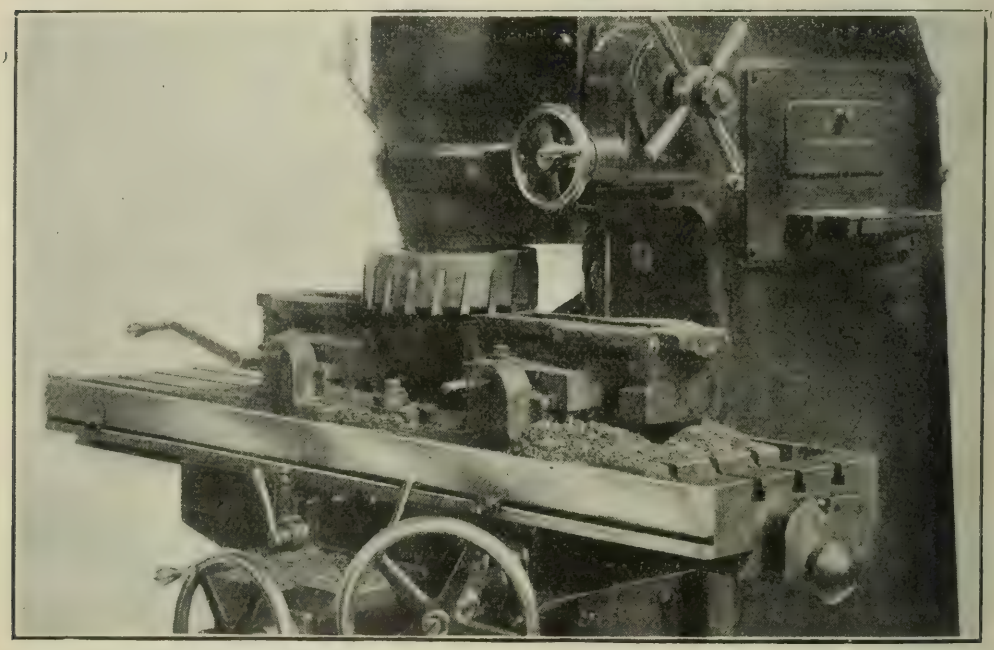


FIG. 25 HIGH-POWER FACE MILL ROUGHING BOTTOMS OF VISE BODIES. ONE PIECE IS CHUCKED WHILE THE OTHER IS MILLING.

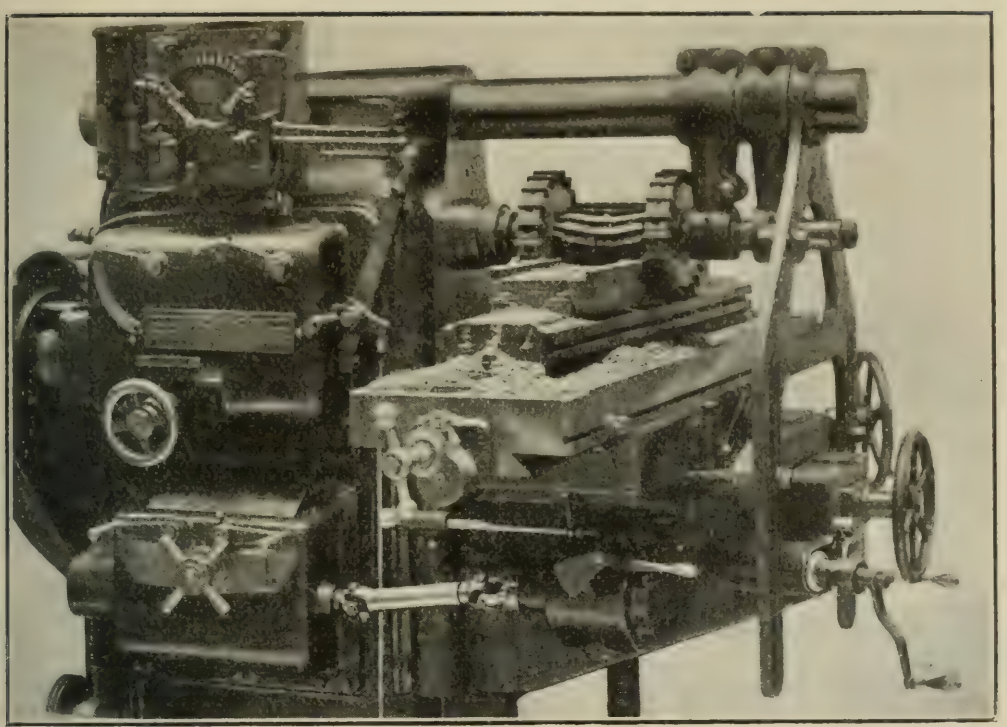


FIG. 26 WIDE-SPACE GANG ROUGHING VISE HOUSINGS

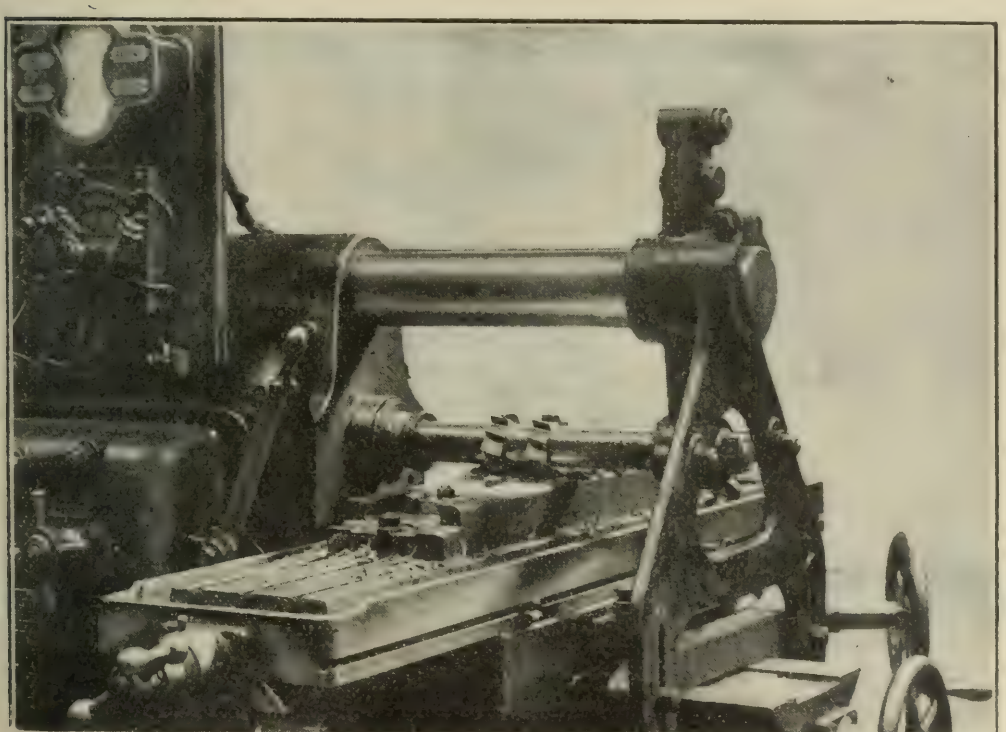


FIG. 27 NEW STYLE GANG AT WORK ON STEEL

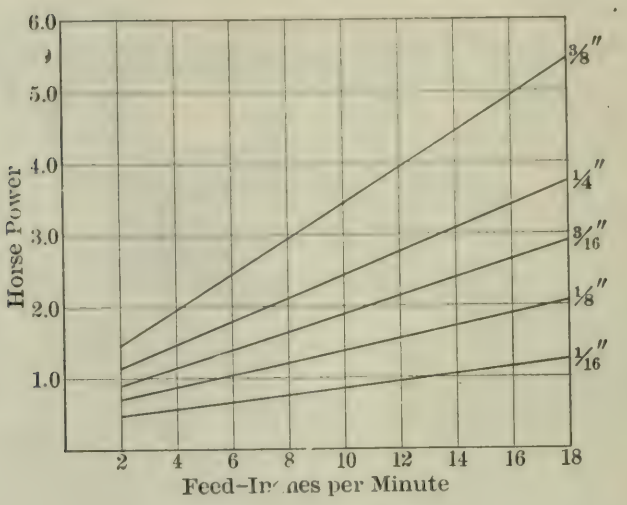


FIG. 28 EFFICIENCY CURVES SHOWN IN FIG. 6 REDUCED TO 1 IN. WIDTH OF CUT

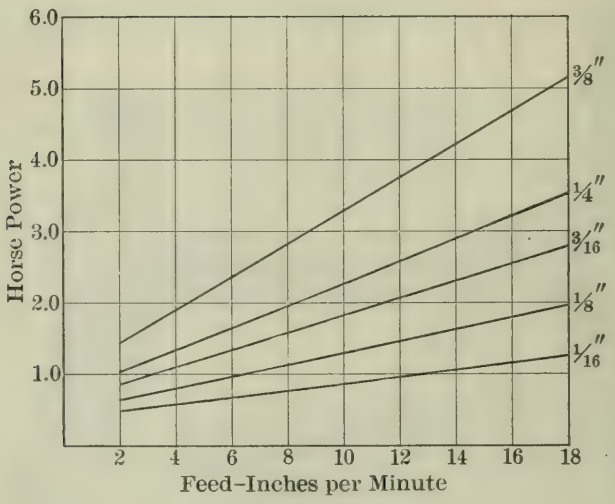


FIG. 29 EFFICIENCY CURVES SHOWN IN FIG. 7 REDUCED TO 1 IN. WIDTH OF CUT

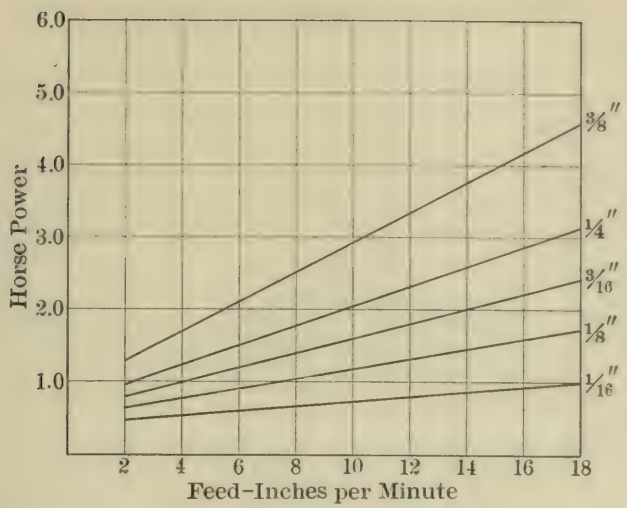


FIG. 30 EFFICIENCY CURVES SHOWN IN FIG. 8 REDUCED TO 1 IN. WIDTH OF CUT

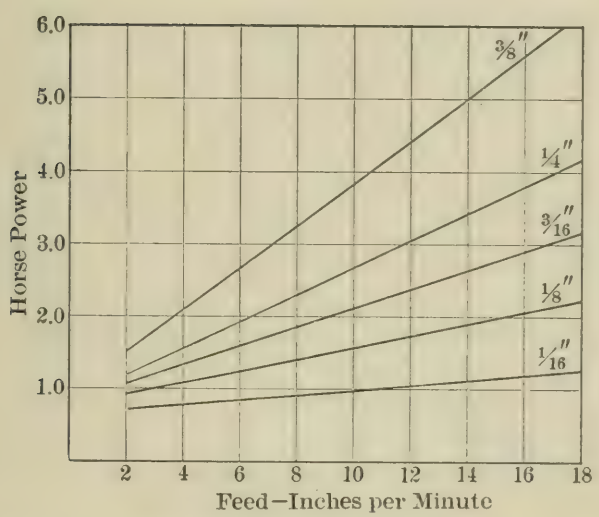


FIG. 31 TESTS OF END MILLING CUTTER 3 IN. DIAMETER, 14 TEETH. CUTTING CAST IRON, CORRECTED FOR HARDNESS. WIDTH OF CUT $2\frac{7}{8}$ IN. REDUCED TO BASIS OF 1 IN.

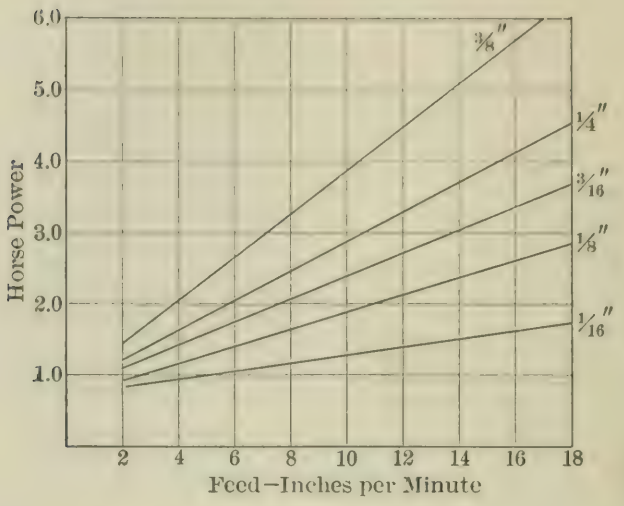


FIG. 32 TESTS OF END MILLING CUTTER 5 IN. DIAMETER, 20 TEETH. CUTTING CAST IRON, CORRECTED FOR HARDNESS. WIDTH OF CUT 3 IN., REDUCED TO BASIS OF 1 IN.

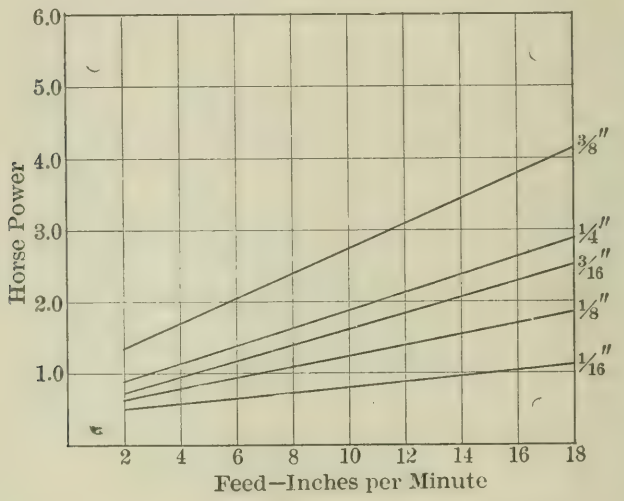


FIG. 33 TESTS OF REGULAR FACE MILLING CUTTER 9½ IN. DIAMETER, 22 TEETH. CUTTING CAST IRON, CORRECTED FOR HARDNESS. WIDTH OF CUT 6 IN., REDUCED TO BASIS OF 1 IN.

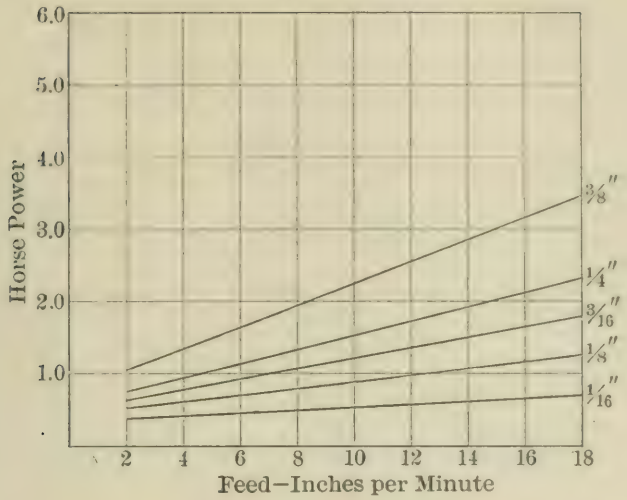


FIG. 34 TESTS OF HIGH-POWER FACE MILLING CUTTER 8 IN. DIAMETER, 6 IN. TEETH. CUTTING CAST IRON, CORRECTED FOR HARDNESS. WIDTH OF CUT 12 IN. REDUCED TO BASIS OF 1 IN.

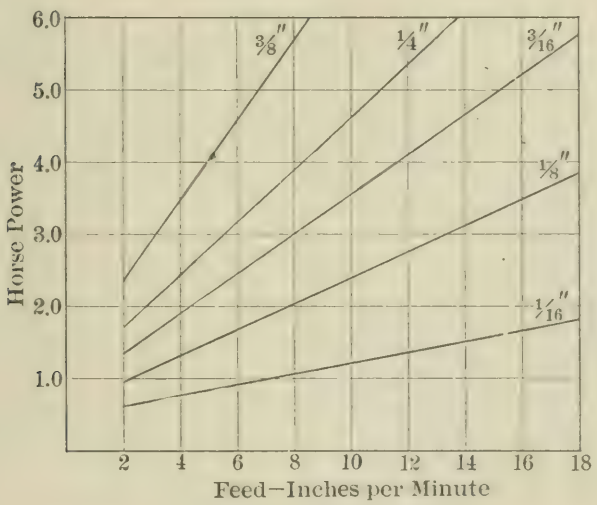


FIG. 35 TESTS OF SPIRAL NICKED MILLING CUTTER 3½ IN. DIAMETER, 14 TEETH, AND ABOUT ¾ IN. PITCH. CUTTING MACHINE STEEL: 0.50 MANGANESE, 0.20 CARBON, 55,000 LB. TENSILE STRENGTH. WIDTH OF CUT 6 IN., REDUCED TO BASIS OF 1 IN.

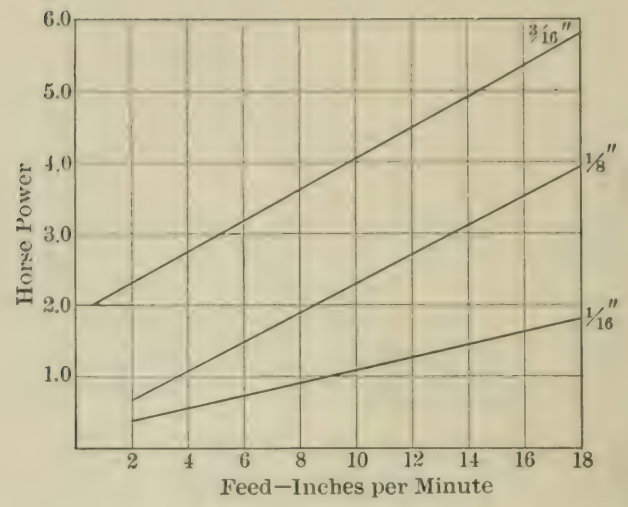


FIG. 36 TESTS OF REGULAR FACE MILLING CUTTER 9½ IN. DIAMETER, 22 TEETH. CUTTING MACHINE STEEL; 0.50 MANGANESE, 0.20 CARBON, 55,000 LB. TENSILE STRENGTH. WIDTH OF CUT 6 IN., REDUCED TO BASIS OF 1 IN.

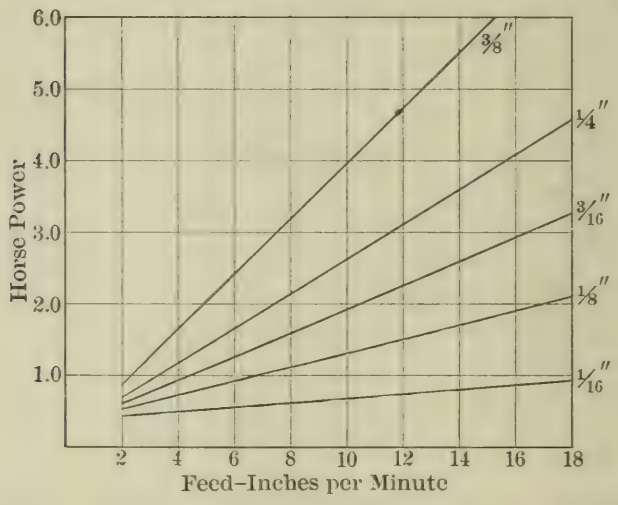


FIG. 37 TESTS OF HIGH-POWER FACE MILLING CUTTER 8 IN. DIAMETER, 12 TEETH. CUTTING MACHINE STEEL: 0.50 MANGANESE, 0.20 CARBON, 55,000 LB. TENSILE STRENGTH. WIDTH OF CUT 6 IN., REDUCED TO BASIS OF 1 IN.

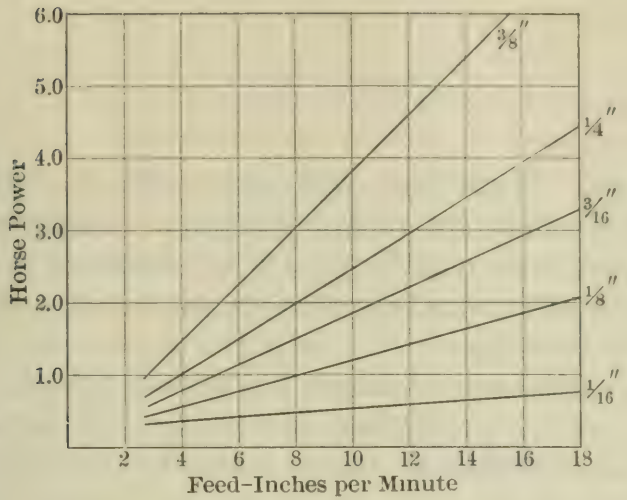


FIG. 38 TESTS OF HELICAL MILLING CUTTER $3\frac{1}{2}$ IN. DIAMETER, $4\frac{1}{4}$ IN. LEAD, $2\frac{1}{8}$ IN. PITCH. CUTTING MACHINE STEEL: 0.50 MANGANESE, 0.20 CARBON, 55,000 LB. TENSILE STRENGTH. WIDTH OF CUT $5\frac{1}{2}$ IN., REDUCED TO BASIS OF 1 IN.

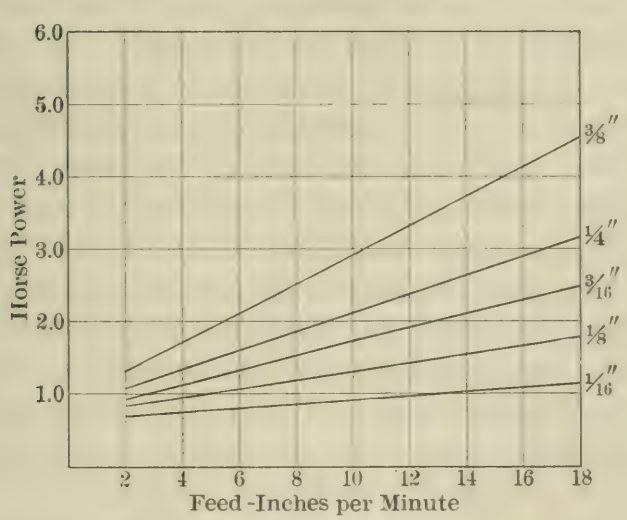


FIG. 39 TESTS OF HELICAL MILLING CUTTER $3\frac{7}{16}$ IN. DIAMETER, $4\frac{1}{4}$ IN. LEAD, $\frac{15}{16}$ IN. PITCH. CUTTING CAST IRON, CORRECTED FOR HARDNESS. WIDTH OF CUT 6 IN., REDUCED TO BASIS OF 1 IN.

DISCUSSION

JOHN PARKER. Much study and many years of actual experience have been involved in the present design of cutters, and any type that possesses any one quality to such an abnormal extent that its other attributes are made less efficient cannot be accepted as the ideal cutter for general use.

The type of cutter Mr. DeLeeuw has introduced has undoubtedly the merit of consuming less power than the regular cutter, as shown by his tests and confirmed to a large extent by a number of comparative tests made at the works of the Brown & Sharpe Manufacturing Company. This distinctive quality is of interest and gives the cutter a value which would be greater were it not that while for the heavier class of milling machines the new cutter proved entirely capable, on the lighter class satisfactory results were not obtainable.

It is evident that the new cutter, having so few teeth, produces a hammering action on the work greater than that of the standard cutter, because there are not sufficient teeth in contact to steady the action; and, unless the machine has the rigidity and massiveness possessed by the heavier class of machines to withstand this pounding effect, the smooth action, so essential to good milling practice, is not obtained. This undesirable feature was very noticeable on a No. 2 Universal milling machine, which is essentially a tool-room machine and necessarily of a light character to enable it to be handled quickly and easily. The pounding was so severe that the tests had to be suddenly terminated, owing to the abusive action to which the machine was subjected. In another machine, somewhat heavier than the one referred to, work, when using the ordinary cutters, could be held in a vise clamped in the usual way; but, with the new cutters it would be pushed bodily out of the vise, and auxiliary means had to be provided to hold it in place. This would indicate that the new cutters would not be suitable for that class of work, when, for some reason it could not be so firmly secured as desired, or when great care would have to be exercised in clamping down work to the fixture or machine table, to prevent springing.

Comparative tests were made between the special and the standard milling cutters, in both cast iron and steel, on the No. 5 Brown &

Sharpe plain milling machine. The machine was motor-driven and was connected to a Westinghouse graphic meter for recording the tests. This is probably one of the best existing methods for making comparative tests. The cutters used were $3\frac{3}{4}$ in. in diameter, 12 deg. spiral, right- and left-hand. The special cutters had 9 teeth, and the standard 16. These tests showed that, in removing 24 cu. in. of cast iron per minute, the new cutters consumed 20 per cent less power than the standard cutters, 15 per cent in removing 15 cu. in., and 14 per cent in removing 12 cu. in. of machinery steel. As regards the smoothness of action, there was very little difference noted between the two types; both were very good. A finishing cut was taken with each cutter in both cast iron and steel; all the surfaces were excellent, the standard cutter having a trifle better finish.

Comparative tests were made on the No. 3 Brown and Sharpe plain milling machine with both the new and the regular type end mills, of the following diameters and number of teeth: 1 in., 4 and 10; $1\frac{1}{2}$ in., 6 and 12; and 2 in., 8 and 14. In all cases the end mills with the small number of teeth consumed less power in both cast iron and steel. The gain in cast iron was 19 per cent for the 1 in. diameter, 16 per cent for the $1\frac{1}{2}$ in. diameter, and 18 per cent for the 2 in. diameter; and in steel, 14 per cent for the 1 in. diameter, 12 per cent for the $1\frac{1}{2}$ in. diameter, and 9 per cent for the 2 in. diameter. In these tests it was found that the new style mill had a tendency to work loose, notwithstanding that the hand of the spiral in relation to the rotation of the mill, had the effect of pushing it in towards the spindle. This trouble was undoubtedly caused by the fact that the mill had so few teeth that the work imposed upon each tooth was sufficient to produce a pounding effect which jarred the mill loose. It is only fair to say that the cuts taken were far beyond what would be used in ordinary practice, and that possibly under ordinary working conditions no bad effect might result. However this may be, this point would have to be carefully tried out before this type of cutter could be accepted in place of the regular, which developed no such trouble.

In making the tests on the No. 5 plain milling machine with the milling cutters, the point regarding lack of chip room that Mr. DeLeeuw emphasizes was closely watched in the standard cutters, and, although the chip taken was far in excess of what would be demanded in ordinary practice, namely, $\frac{1}{10}$ in. deep and 10 in. feed per minute, there was apparently sufficient space, as not the slightest sign of clogging was in evidence.

For some time past, stock face milling cutters have been made with the teeth set at an angle, both on the face and periphery, whereas the older type had radial teeth, both on the face and periphery; in both cases the pitch of teeth approximates $1\frac{1}{2}$ in. As a matter of interest, comparative tests were made. The results were, with the new type, a saving of 12 per cent of power in cast iron, and a gain of 6 per cent in favor of the old type in steel. This latter was somewhat surprising; and, were it not for the very careful manner in which these comparative tests were made and recorded, the result might be questioned.

A test was made in steel with the new and old side milling cutters, 4 in. in diameter, 14 and 28 teeth respectively. One pair of each was used. A gain in power of 20 per cent in favor of the new was obtained. In this test the cutters with the smaller number of teeth cut more easily and the work had a better finish.

In briefly summing up the situation, I believe Mr. DeLeeuw has developed an interesting type of cutter which possesses to a marked degree a valuable attribute, that of saving power; and, in certain classes of work, such as heavy manufacturing milling, it will doubtless find a field to which its characteristics are most suitable. But, in view of the knowledge gained by the comparative tests made, I doubt very much whether it will be as successful a cutter for general purposes as its predecessors.

A. F. Murray. The results of Mr. DeLeeuw's experiments are in accordance with theories which I have been advocating for several years. At the Blake & Knowles Steam Pump Works we are continually re-cutting old milling cutters for our manufacturing departments, and invariably reduce the number of teeth about half, setting the mill deep enough to cut out completely every other tooth. In most cases we have done this with standard bevel cutters for milling the teeth. We are now glad to avail ourselves of the results of Mr. DeLeeuw's experiments as to changes in these cutting angles for end and spiral mills.

We have an equipment of the high power face mills, shown in Fig. 16, and we have recently had some new high-duty 16-in. and 20-in. face mills made for a heavy 4-spindle milling machine. Fig. 40 shows the two 16-in. cutter heads at work with the two 20-in. cutter heads standing alongside the machine. An old pattern 12-in. cutter head is shown on the horizontal spindle.

The method of attaching these cutters to the spindle is worth noting. The spindle has a taper hole in which is fitted a short arbor 3 in. in diameter used only for centering the cutter. The spindle also has a flanged end 7 in. in diameter and a key 1 in. wide permanently dovetailed across its face. The flange is also tapped for four

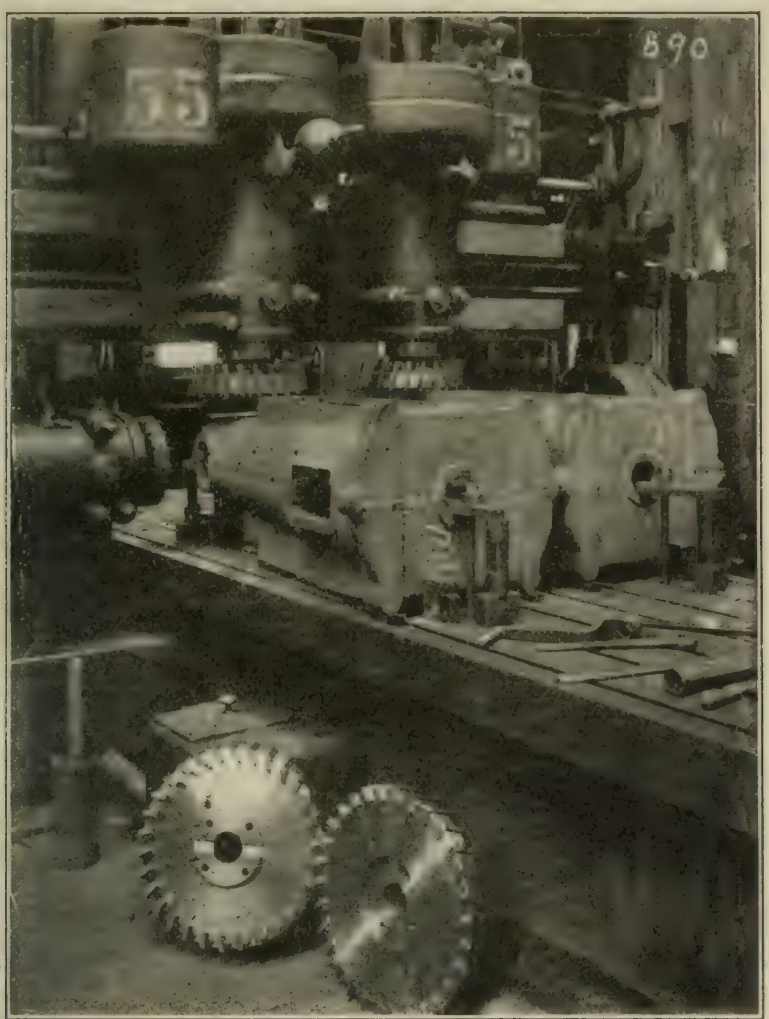


FIG 40 TWO 16-IN. CUTTER HEADS AT WORK

$\frac{3}{4}$-in. bolts which hold the cutter firmly and squarely against the face of the spindle but do not assist in driving, as they go through clearance holes in the cutter head. This holding means was suggested by one of the cutter manufacturers and adopted after a careful trial of the cutters screwed on the end of the spindle and several arrangements with drawback bolts and with both face and longitudinal keys. Fig.

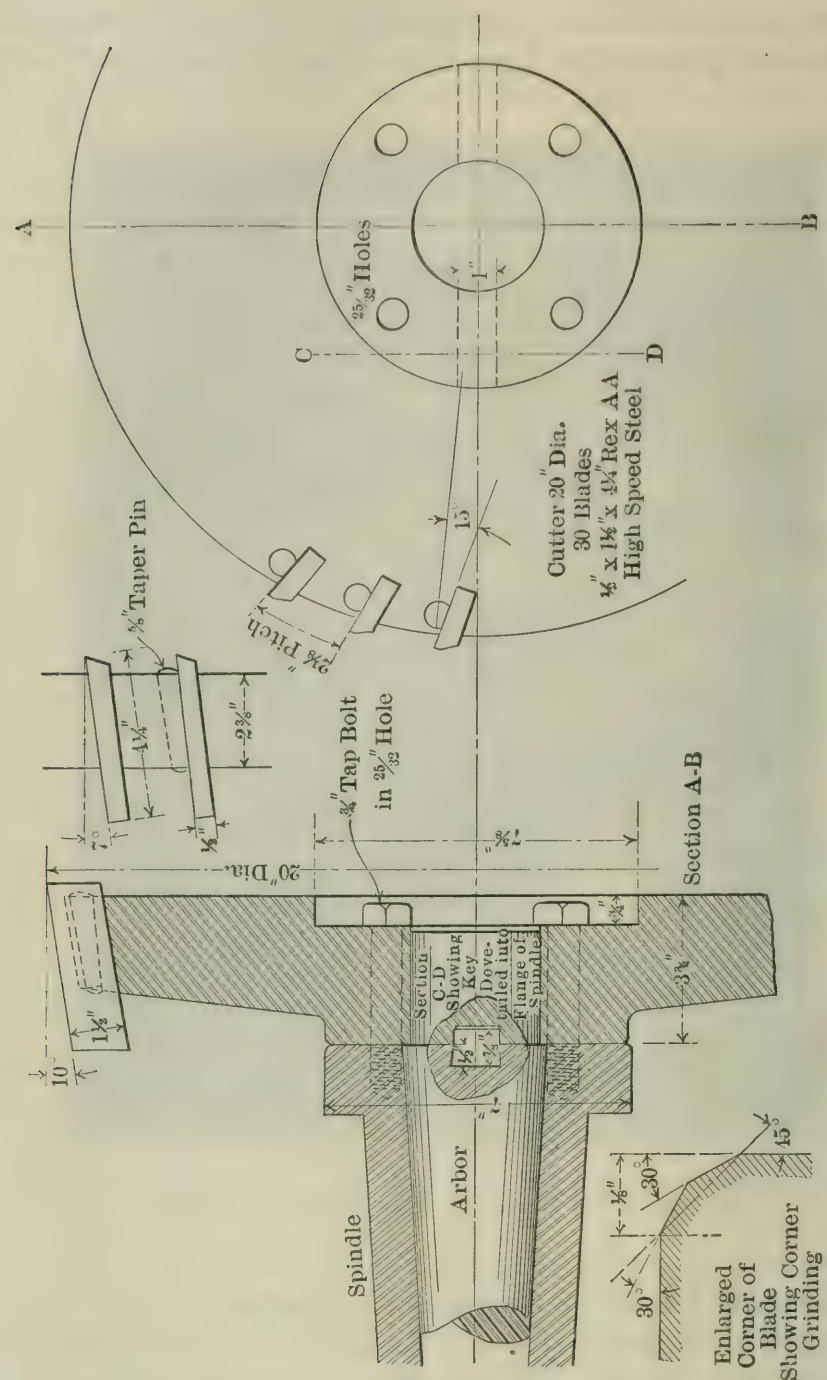


FIG. 41 CONSTRUCTION OF CUTTER HEADS

41 shows the construction of these cutters. It will be noted that the bodies are shaped like the frustum of a cone with the blades pitched back at an angle of 10 deg. and projecting beyond the rear of the body as much as the design permits. In this way long life of cutter blades is secured with a minimum amount of grinding. The other clearance angles, it will be noted, are practically the same as in Fig. 16. Instead of three settings in grinding required for cornering the blade, we have obtained good results with two settings as shown in detail in Fig. 41.

It is our intention to construct experimental cutters with coarser pitch of blades than those we have at present ($2\frac{1}{2}$ in. and 3 in. circu-

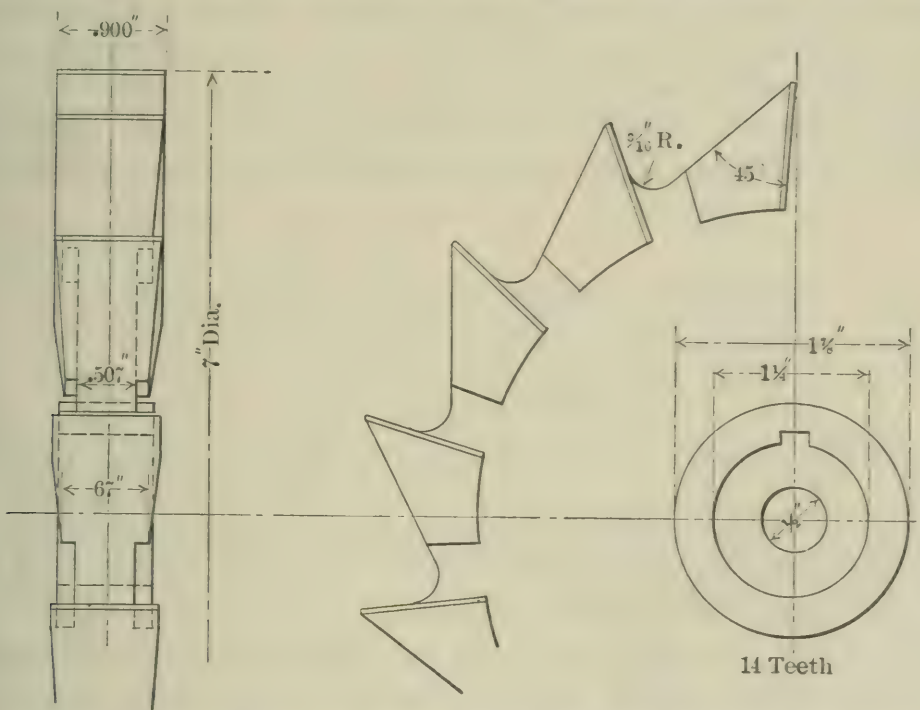


FIG. 42 CUTTER USED ON HEAVY TYPE LINCOLN MILLING MACHINE

lar pitch instead of $2\frac{1}{8}$ in.), as we do not believe we have reached the limit in this direction.

Referring to Mr. DeLeeuw's remarks on end mills, results obtained in a splining machine recently put on the market, by the use of fish-tail cutters running at high speed with ample lubrication, have served to indicate that a reduction in the number of teeth of end mills was desirable. We have recently put into use several 3-in. and 4-in. inserted blade end mills with 6 and 8 teeth respectively, the blades being constructed of $\frac{3}{8}$ by $1\frac{1}{16}$ in. high-speed steel, raked in the same manner as the larger face mills shown in Fig. 16.

Saws, slotting and side mills, as ordinarily made, have too many teeth. It is my opinion that a large proportion of the success obtained by the use of inserted tooth mills for this purpose has been due to the enforced reduction in the number of teeth. Several years ago, while connected with a small arms factory, I assisted in some experiments which were made for increasing the production of a very heavy slotting operation in which a 7-in. cutter, 0.900 in. wide with a $1\frac{1}{4}$ in. hole, buried itself to the hub in a steel forging set at an angle of 45 deg. The operation was performed on a heavy type of Lincoln milling machine. The first cutter employed was a high-speed side mill of regular pattern, which gave very unsatisfactory results and stalled the machine with light feed, choking with chips, although the cut was flushed with a heavy stream of soda water. Some one suggested the cutter shown in Fig. 42 which practically doubled the production, but the output was still below requirements. Opposite corners of alternate teeth were then removed, as shown in Fig. 43.

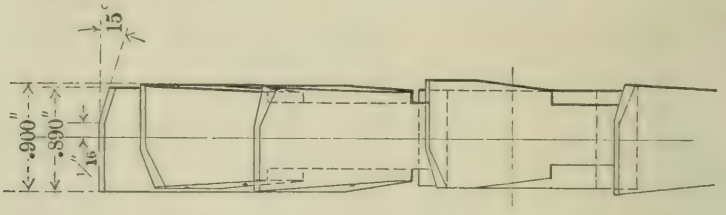


FIG. 43 CUTTER WITH OPPOSITE CORNERS OF ALTERNATE TEETH REMOVED

This slight change enabled us to feed from 1 to $1\frac{1}{4}$ in. a minute as against about $\frac{5}{8}$ in. with the cutter (Fig. 42) and about $\frac{3}{8}$ in. with the standard form of side milling cutter.

About two years ago I repeated this experiment with an inserted tooth side milling cutter 10 in. in diameter, $1\frac{1}{2}$ in. face, 2 in. hole. The pitch of the blades was about the same as that shown in Fig. 17 (being a side mill the blades were set in parallel to the axis), approximately 1 in. circular pitch. This cutter was being used to mill a slot $3\frac{1}{2}$ in. deep by $1\frac{3}{4}$ in. wide in U. S. Navy Class *A* steel forgings, having a tensile strength of 80,000 lb. per sq. in.

The cutter, as furnished by the manufacturer for straddle milling work, was unable to cut through this material more than $\frac{3}{8}$ in. per min. and the machine could be heard all over the room. This recalled to my mind the former experience above referred to and alternate corners of the inserted blades were ground off at an angle of 15 deg., $\frac{1}{16}$ in. beyond the center, and they were sufficiently offset to enable the $1\frac{1}{2}$ in. cutter to machine the $1\frac{3}{4}$ in. slot to size at only one

cut. This change increased the feed possible from $\frac{3}{8}$ in. up to $1\frac{1}{8}$ in. per min. and gave 200 per cent increase in efficiency by making the chip three times as thick and half as wide as at first and providing additional chip clearance. It is believed that some of the side mills shown in Figs. 14a and b and Fig. 15 would be improved by the adoption of this staggered tooth when used for cutting slots.

The practice above referred to has been used quite frequently in connection with small tee-slotting cutters, but I have never seen it used elsewhere on the larger types of slotting cutters.

W. S. Huson. From the point of view of product, the operation of cutters is paradoxical. In a gear-cutting machine, we have not been able to get the same results with the wide-spaced edges of the standard cutter of today as with cutters having several times as many cutting edges. Gear teeth are smoother and run better, though the cutter is much more expensive to maintain. Even after all the investigation by prominent cutter people, the larger number of teeth in a milling cutter gives better gears.

We get better results in surface milling from wide-spaced cutting edges than from the usual close-spaced. We have found in testing work milled with cutters having many teeth that when the cut was finished, say at 3 p.m., and tried with straight edge and gages, it was right, while the next morning it was out. There are two causes for this: the peening effect of the cutter, and another which I think is not fully considered, but which I believe is borne out by experience, namely, that the teeth of a milling cutter punch or force little particles of cutting dust into the interstices or pores of the iron, which finally respond to the force exerted on them, and throw the work out of alignment. A cutter with fewer cutting edges for surface milling does not give a bright finish, but the product requires less subsequent filing and fitting. It is for this reason that the question of milling cutters is paradoxical, for in the case of gear cutters with many cutting edges we get a smoother gear tooth, whereas in surface milling we get a truer surface with fewer edges.

I agree with the author that in surface milling the fewer cutting edges, the more chip clearance, and hence the more permanent, if not quite so smooth, work will give better final results than cutters with close-spaced edges. The amount of power consumed is of little importance in the final cost, if as a result of rapid machine output, manual labor must be used to make the work acceptable.

P. V. VERNON. An article published by the writer in the Engineer[1] is in substantial agreement with Mr. DeLeeuw. It emphasizes the superiority of coarse pitch over fine pitch cutters, with a diagram (Fig. 2, see Fig. 44) showing the way in which the chip is formed, similar

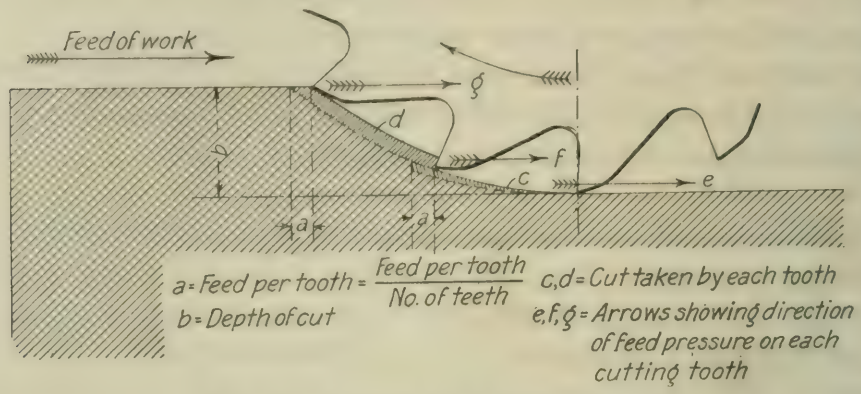


FIG. 44 ACTION OF A SIDE CUTTER

to the method described by Mr. DeLeeuw. Fig. 3 (see Fig. 45) shows the way the chip is formed by a face mill. In a table compiled for the article, the same number of teeth for roughing cutters is recommended.

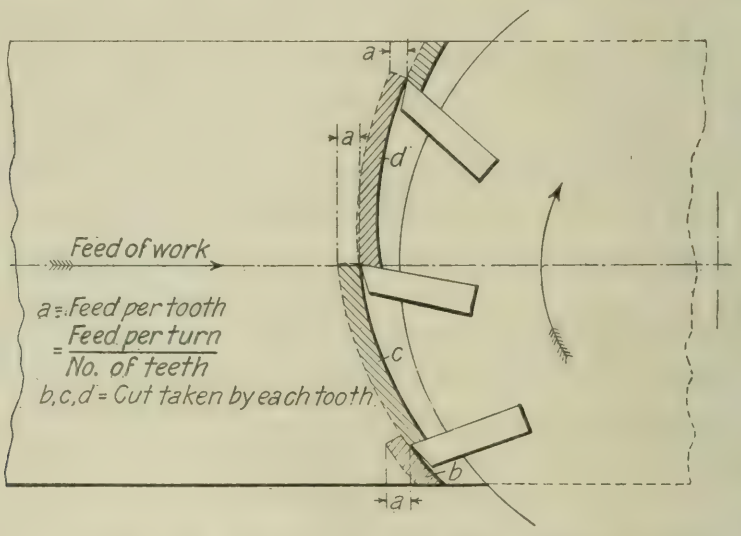


FIG. 45 ACTION OF A FACE CUTTER

Fig. 4 of the article (see Fig. 46) shows an example of coarse pitch milling and Figs. 47 and 48 herewith illustrate the heavy milling per-

[1] May 31, 1907.

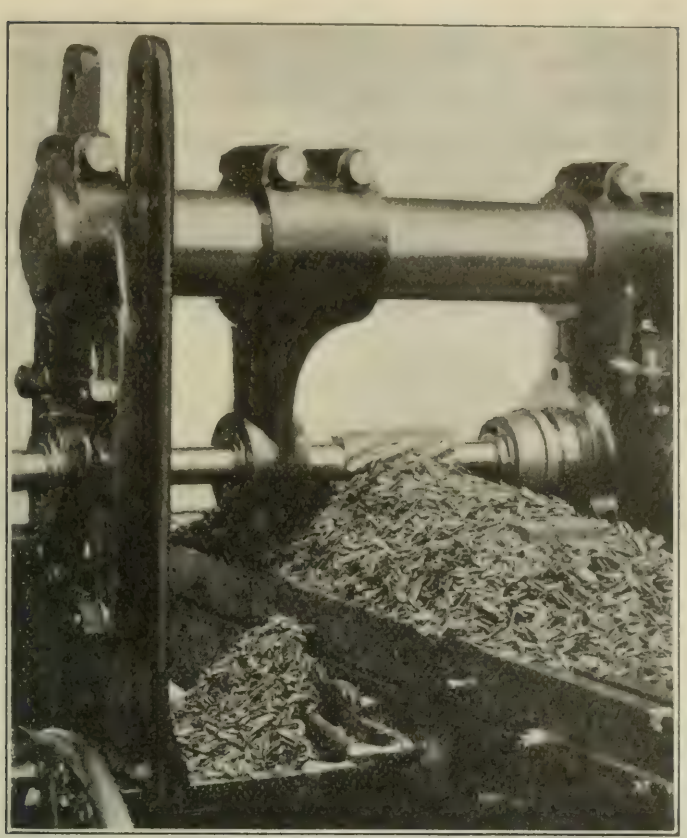


FIG. 46 EXAMPLE OF COARSE PITCH MILLING

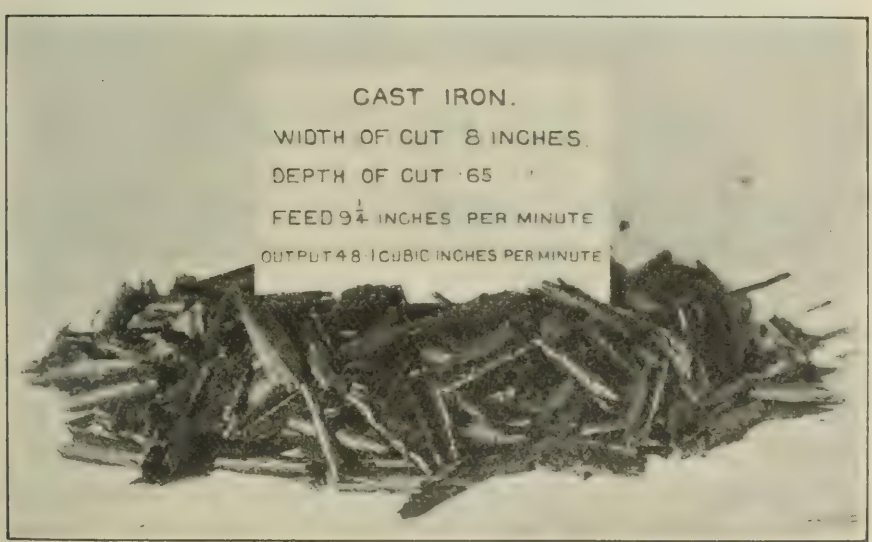


FIG. 47 EXAMPLE OF HEAVY MILLING PERFORMED BY NO. 22 SINGLE PULLEY PLAIN MILLING MACHINE

formed by the No. 22 single pulley plain milling machine of Alfred Herbert, Ltd. This machine corresponds approximately in size to the No. 4 machine of the usual American specifications. The cuts in question were taken with coarse pitch cutters, $4\frac{1}{2}$ in. in diameter, the size of the arbor being 2 in.

On the subject of face mills, I may say that inserted tooth cutters having the same features as those described by Mr. DeLeeuw have been made and sold by this same firm for the past eight years. Fig. 49 shows an early type of cutter on this principle, dating back to 1903, and Figs. 50 and 51 show the present practice. In the latter the cutters are provided with rake in two directions.

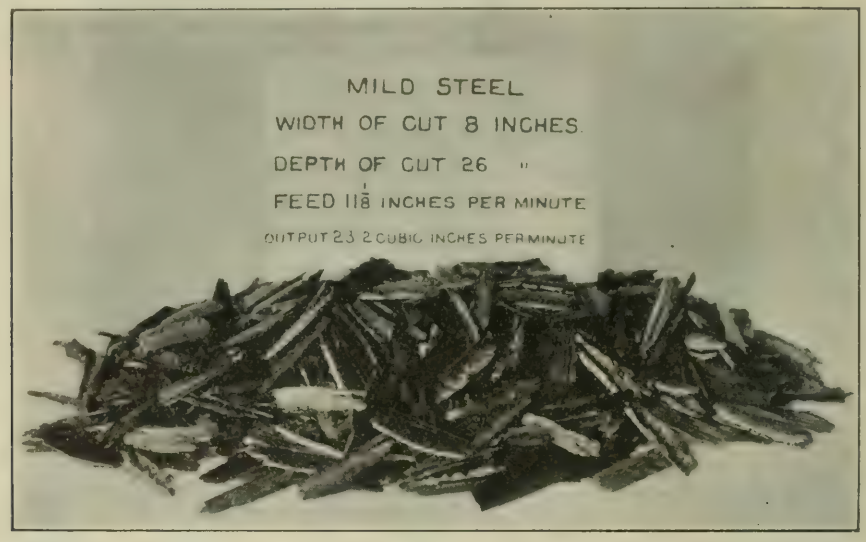


FIG. 48 EXAMPLE OF HEAVY MILLING PERFORMED BY NO. 33 SINGLE PULLEY PLAIN MILLING MACHINE

With regard to notched cutters with the notches relieved on both sides, such cutters were originally introduced into England by William Muir & Company, in 1880. The form of the notch was, however, rectangular instead of V-shaped.

I am not entirely in agreement with the author concerning helical cutters, as I have tried them and although the chips come off in the way stated, yet they are much hotter than with ordinary coarse pitch roughing cutters, owing to the amount of rubbing being so much greater, and the end thrust was such that when used on machines with plain thrust washers great difficulty was experienced in keeping the bearing cool. Unless made right and left hand I do not think such cutters are likely to come into general use.

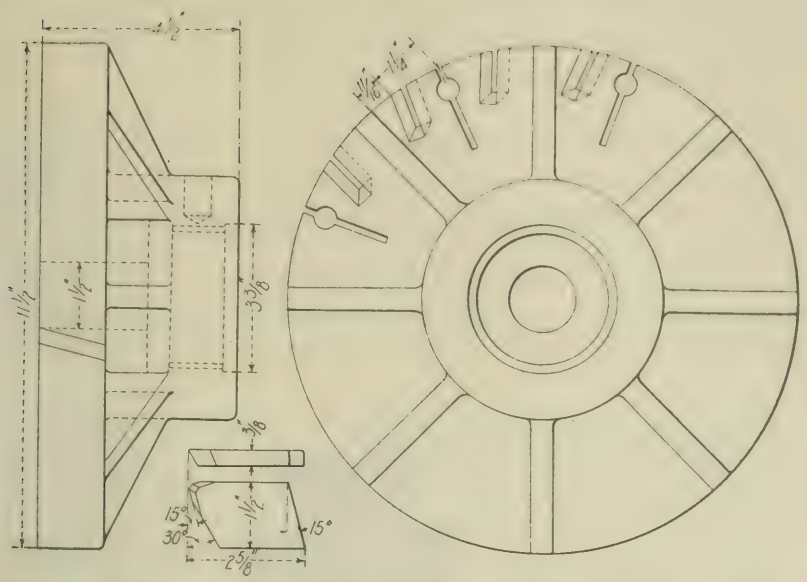


FIG. 49 EARLY INSERTED TOOTH FACE CUTTER, FEBRUARY 1903

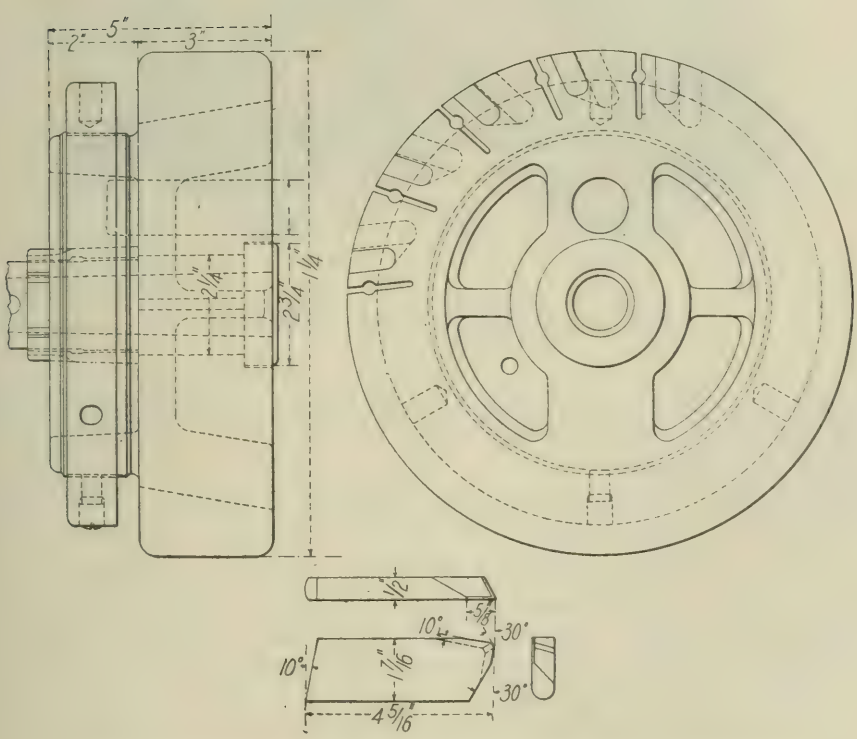


FIG. 50 PRESENT STYLE INSERTED TOOTH FACE CUTTER

[A reference by Mr. Vernon in the article in the Engineer to the use of cutters of coarse pitch is contained in the following extract from the article to which he refers.—EDITOR.]

The usual limitations of output when milling are caused by spring or chatter due to weakness of the work, lack of driving power, weakness of the machine, or of the cutter arbor; weakness of the feed motion; weakness of the body of the cutter, not of the teeth; or insufficient room for chips. In order to get the maximum amount of work then from each tooth of the cutter it is necessary to give the following requirements, assuming that the work is massive enough to stand the heaviest cut that will be taken: (*a*) sufficient driving power; (*b*) a strong

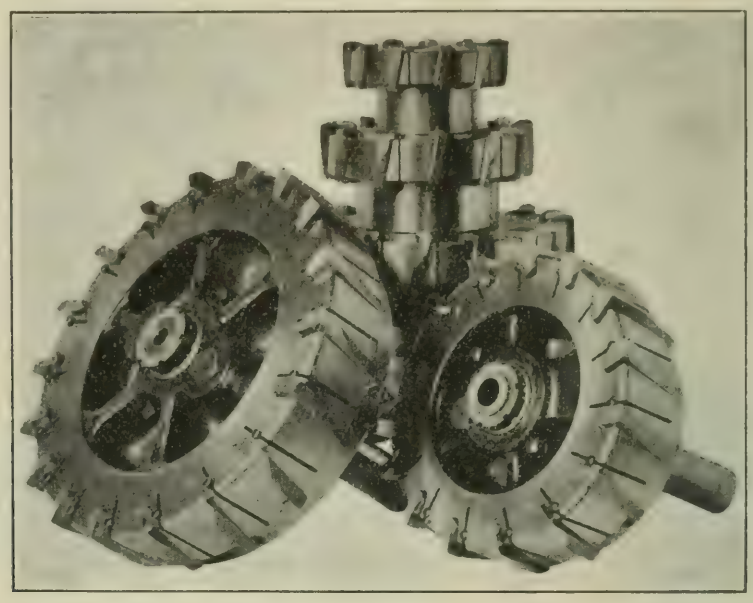


FIG. 51 INSERTED TOOTH CUTTERS

enough machine; (*c*) a stiff enough arbor; (*d*) a regular and powerful feed; (*e*) enough metal in the body of the cutter; (*f*) ample room for chips; in other words, teeth of coarse pitch.

A milling cutter will do more work when each tooth gets well under the surface; in other words, when the feed per tooth is sufficient to enable a cut rather than a scrape to be taken. With coarse pitch cutters less teeth are cutting at one time, and a given feed power is more effective on each tooth than in the case of fine pitch cutters. There is also more room for chips. The space per tooth available for chips varies approximately as the square of the pitch of the teeth, and the total space for chips all round the cutter varies directly as the pitch. Coarse pitch cutters can also have stronger teeth than fine pitch cutters, as they permit of a wider land behind the cutting edge, without unduly robbing the space available for chips, and incidentally the thicker tooth is not so easily affected by heat generated while cutting. In addition to this, the thicker chips taken by coarse pitch teeth occupy less space in proportion to

their weight than thin chips, thus further favouring the coarse pitch cutter. Most of the published tables of milling cutter dimensions specify too many teeth in the cutters, and users of modern machines will be well advised not to rely on such tables, but to favour very much coarser pitches.

Table 1 has been found satisfactory for general use, but is put forward more as a suggestion than with the idea that it is a final statement of ultimate possibilities.

TABLE 1 CYLINDRICAL CUTTERS

Ordinary Cutters for Roughing and Finishing			Cutters for Heavy Roughing only
Diameter		No. of teeth	No. of teeth
⅛	3/16	5	..
¼	3/16	6	..
⅜	1¼	8	..
1⅜	1½	9	6
1⅝	1¾	10	6
2	2⅜	12	7
2½	3⅜	14	8
3½	3¾	16	9
4	6	18	10

With roughing cutters made as per the last column, only one tooth will be cutting at a time on all ordinary roughing work. The spiral must, therefore, be such as to ensure continuity of torque, or the result will be an intermittent cut. A good angle of spiral for general work is 27 deg. with the axis of the cutter. It must not be forgotten that coarse pitch of teeth necessitates short pitch of spiral.

THE AUTHOR. In a general way, the results of Mr. Parker's experiments are in line with mine and perhaps the greatest difference between them is a matter of amount. The difference between the power consumed with the old and new style cutters I found to be greater than that obtained by Mr. Parker, and further experiments which I have made since writing this paper show even greater differences than were found at first, due to the fact that they were made with cutters with a wider spacing of the teeth.

As to the hammering effect produced by the wide-spaced cutter, I found this to exist only with roughing cuts of very moderate depth and it was overcome entirely by making the spiral steeper than 10 deg. Experiments carried out with cutters with 20-deg. spiral were entirely successful and it was found impossible to produce a set of conditions in which this cutter would cause hammering. At the same time, the end thrust was so slight, that its effect could not be noticed.

Mr. Parker's method of using a Westinghouse graphic meter for recording the tests raises doubts in my mind as to whether it is possible to get a simultaneous reading of power consumed and speed of machine, though, of course, I have no reason to doubt that Mr. Parker has taken care of this feature in some way. If so, it would be interesting to know in detail his method of testing.

The results of his tests of wide and narrow spaced face mills, (the wide spaced mills having double rake), are widely at variance with my results; and if I had not repeated carefully these tests a great number of times, I might be tempted to repeat them again. It is probable that there are some essential constructive differences between the cutters used by Mr. Parker and by myself, which may explain this great difference in results.

The only thing I want to add to Mr. Murray's remark is that there seem to be some points in the construction of this cutter which I would consider improvements, especially the cutting away of one side of the alternate teeth.

In regard to Mr. Huson's remark that "the amount of power consumed is of little importance in the final cost, if as a result of rapid machine output, manual labor must be used to make the work acceptable," I would not wish anyone to think that I have advocated rapid removal of metal at the expense of finish, nor that these new cutters produce such an undesirable result. On the contrary, I found less disturbance of flatness and straightness, on account of pressure and heat. Another angle of this question is that it is not so desirable to use less power because power costs money, but because all power used which is not needed directly for separating the chip from the work, is power used to dull or break cutters, to distort work and to wear or ruin the machine, and if I would get a bonus for every horsepower used on machine tools, I would still aim to make the machines in the shop do their work with the smallest possible amount of power.

No. 1314

TOPICAL DISCUSSION ON LARGE BLAST-FURNACE GAS-POWER PLANTS

A. E. Maccoun[1] said that the cost of a gas engine is very much higher and the space required very much larger than that for a steam engine of the same power. Some have tried to overcome this difficulty by running the engine at excessively high speeds, but this practice cannot be recommended. With large gas engines, for instance an engine with a cylinder 48 in. in diameter, it becomes nearly impossible to design the working parts, such as cylinders, piston rods, pistons, frames, crank pins, etc., sufficiently large and strong and to find material good enough to withstand the enormous strains to which they are subjected. In large gas engines the parts are continually subjected to fatigue on account of the high temperature and pressure conditions, from which many cracked pistons, rods, heads and cylinders have resulted, and much of the economy gained in the use of fuel has been lost in the enormous repairs required on many of these engines.

So far as reliability is concerned, although they may require more skilled supervision, excellent results are being obtained at many plants, the record of operation with the percentage of time operated during the year at the Edgar Thomson furnaces being from 75.87 to 90.3 per cent. The most important part of a gas engine is the cylinder and the troubles from this have not yet been completely overcome. The greatest difficulty is due to cylinders cracking, thus allowing the water to leak into them, seriously interfering with the operation of the engines. To prevent this, great care must be taken to provide for the extreme temperature and pressure conditions and the cylinders necessarily have to be very carefully water-jacketed to prevent unequal expansion.

[1] Superintendent of Furnaces, Edgar Thomson Steel Works, Braddock, Pa.

In abstract form. Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers. A more complete account appears in The Journal, September 1911.

Iron and steel castings have been used for these cylinders and there is still a difference of opinion as to which is preferable, the speaker favoring the steel casting on account of its greater strength together with its thinner walls, permitting better cooling and helping to prevent the fatigue of the metal. But even this does not eliminate cracking. Large gas-engine cylinder castings should be made as plain and simple as possible, and preferably in halves and bushed. The bushing renders a new cylinder wall available at a cost much less than boring and fitting new pistons, from time to time. The bushing should be easily removable and not require the removal of the cylinder from the engine. Both cast-iron and cast-steel bushings have given satisfaction.

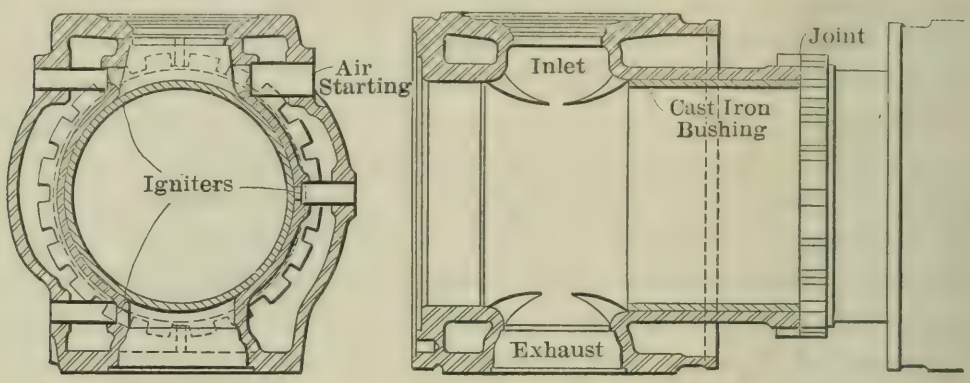


FIG. 1 CROSS-SECTION AND LONGITUDINAL SECTION OF CYLINDER

As large a space as possible should be allowed for circulating water and special attention should be given to the bushings connecting the inner and outer walls of cylinders for igniters, air valves, etc., so they will remain tight and allow for the difference in expansion between the inner and outer walls of cylinders. The limit of size of gas engine cylinders is very uncertain and it looks as if 44 in. in diameter should be about the maximum until further results are obtained from the numerous designs now being tried out. Fig. 1 shows a cast-steel cylinder used on Westinghouse gas engines at the Edgar Thomson Works and Figs. 2 and 3 sectional details.

Piston rods should be made in one piece and should be free from all parts that may become loose. The number of rings in various gas-engine pistons varies from three to eight, from snap rings to sectional rings with keepers, the speaker preferring not to exceed four rings, of the latter type, with good depth and wearing surfaces. The rings should be dowelled in place so that no dowels can come out

and cut the cylinders. It is a most difficult problem to design piston rods strong enough to stand the constant stresses and strains to which they are subjected on large gas engines. The maximum stress on

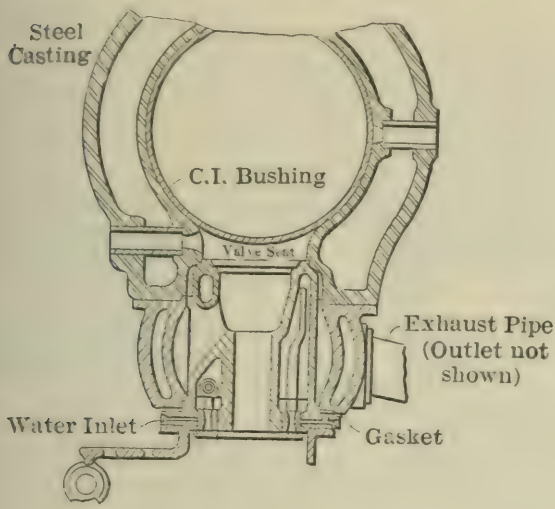


FIG. 2 CROSS-SECTION OF CYLINDER THROUGH IGNITER HOLES

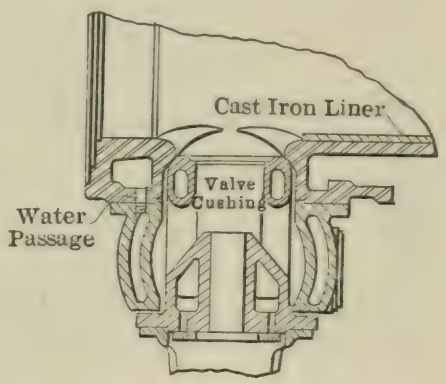


FIG. 3 SECTION THROUGH EXHAUST VALVE

a section of rod due to the explosion pressure, varies from about 6000 to 11,000 lb. per sq. in., on the various American types of large gas engines, and still many of these rods have failed and it is nearly

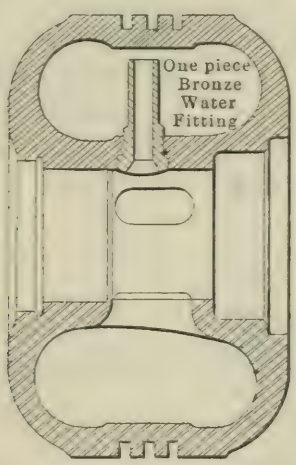


FIG. 4 SECTION OF GAS-ENGINE PISTON; RINGS 1 IN. BY $1\frac{3}{8}$ IN.

impossible to increase the size to any great extent. Some are as large as 13 in. in diameter, but changes that have helped to strengthen them to some extent have been made in the design and material.

There are two general methods of fastening the rods to the cross-heads: by keys through rod, and by thread and nut, or clamp over the threads. Both keys and nuts have given considerable trouble

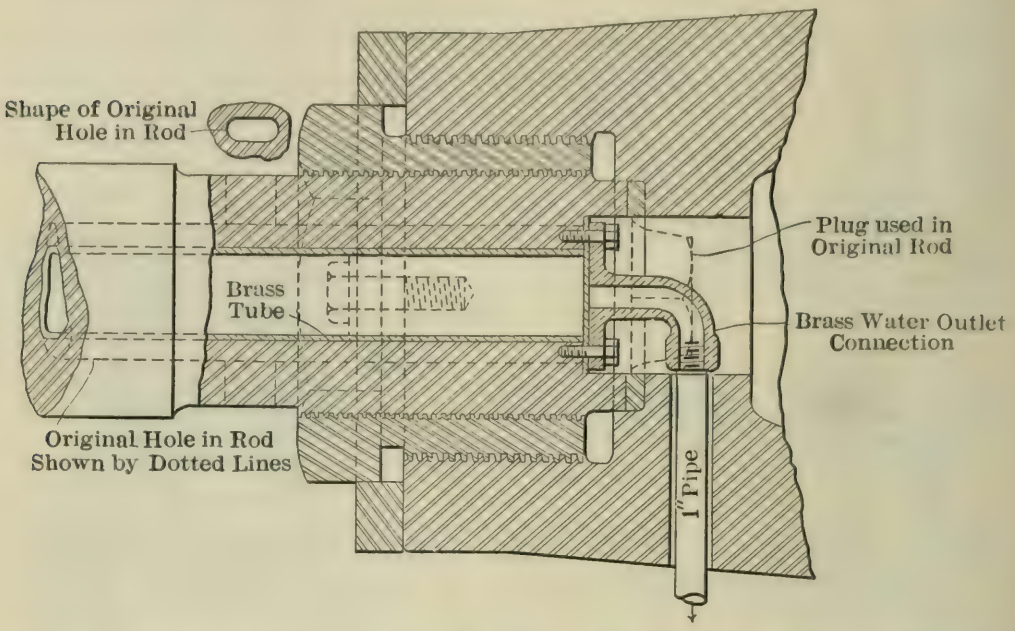

Shape of Original Hole in Rod
Brass Tube
Original Hole in Rod Shown by Dotted Lines
Plug used in Original Rod
Brass Water Outlet Connection
1" Pipe

FIG. 5 SECTION SHOWING PISTON WATER OUTLET AND MAIN CROSS-HEAD CONNECTION

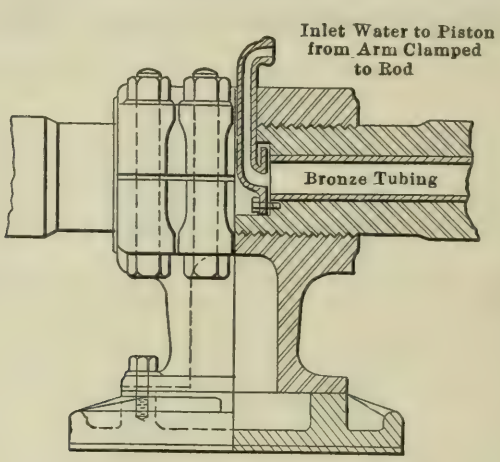


FIG. 6 SECTION SHOWING WATER INLET TO PISTON

and it is hard to say which is the better type to use. The clamp over a thread, or other recess, seems to be preferable, as it does not interfere with the passage of the water from the end of the rod.

The material from which rods were first made was nickel steel, but it was found very unreliable for this particular class of work.

Open-hearth steel of the following composition was found to be more satisfactory:

	Per cent
Carbon	0.45 to 0.60
Manganese	0.45 to 0.60
Phosphorus	under 0.04
Sulphur	under 0.04
Silicon	0.10 to 0.20

The steel is heat treated and has the following physical properties: 50,000 lb. per sq. in. elastic limit; 95,000 lb. per sq. in. ultimate strength; 12 per cent elongation in 2 in. No trouble has been experienced by the wearing of piston rods made from this material, on account of their softness.

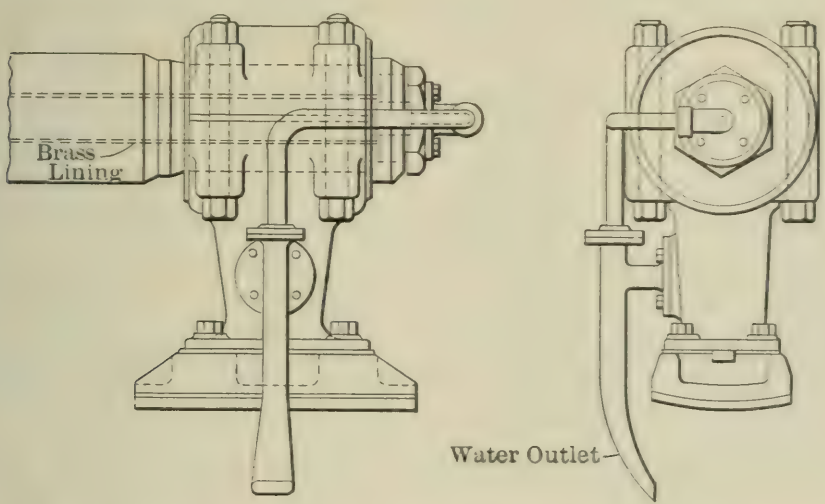


FIG. 7 VIEW SHOWING WATER CONNECTION TO TAIL ROD

The water is brought into and taken out of the ends of the piston rods, there being no holes through the walls of the rods to form places for cracks to start.

In piston-rod packing the most important points to be watched are the fire rings; they should be solid and there should be a sufficient number to reduce the explosion pressure before it reaches the packing rings. It is necessary to have sufficient side clearance so the rings can float. For this 0.005 in. is a fair allowance.

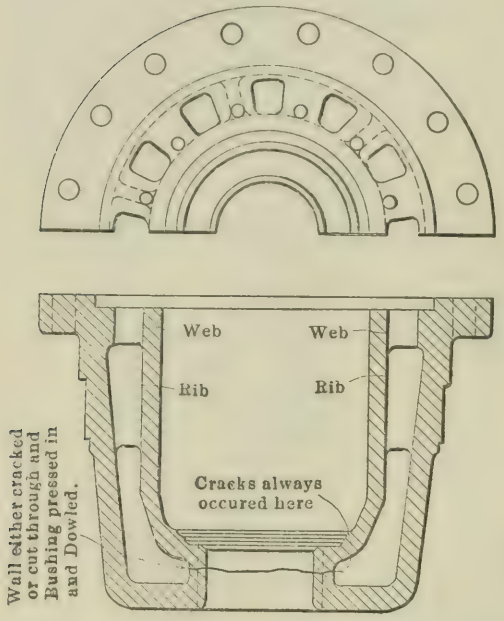
There are still many improvements that can be made on the inlet valve gears of all blast-furnace gas engines. The essentials are as follows:

a A reasonable percentage of dirt in the gas should not affect their operation.

b The wear on all parts, cams, valves, seats, etc., should be easily compensated for in their method of adjustment.

c Ample surfaces should be allowed for all wearing parts, to prevent wear and distortion of the valve setting.

d The adjustment of each valve on the engine should be made so as to be independent of the others.



[FIG. 8 SECTION OF ORIGINAL CYLINDER HEAD

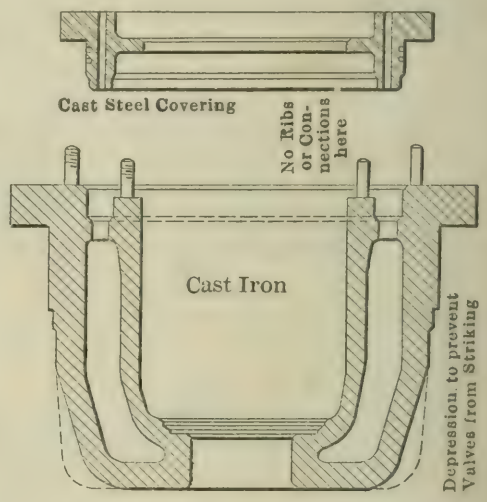


FIG. 9 SECTION OF CYLINDER HEAD OF NEW DESIGN

e The valve gear should also be arranged so that the relative ratio of the air and gas ports can be changed either individually or collectively and very quickly to suit sudden changes in the composition of the gas.

A great many of the gears in use fulfil most of these conditions, but none of them, as far as the speaker is aware, satisfy the last, which is a most important one.

Of the two general systems in use in designing valve gears, constant mixture and constant compression, the speaker found the former better for all load conditions. The oil relay governor was found most satisfactory for operating inlet valves, on account of the large amount of work to be done. The exhaust valves have given very little trouble and do not have to be ground in very frequently. Engine

details illustrating further the practice of the Edgar Thomson Works are shown in Figs. 4 to 10.

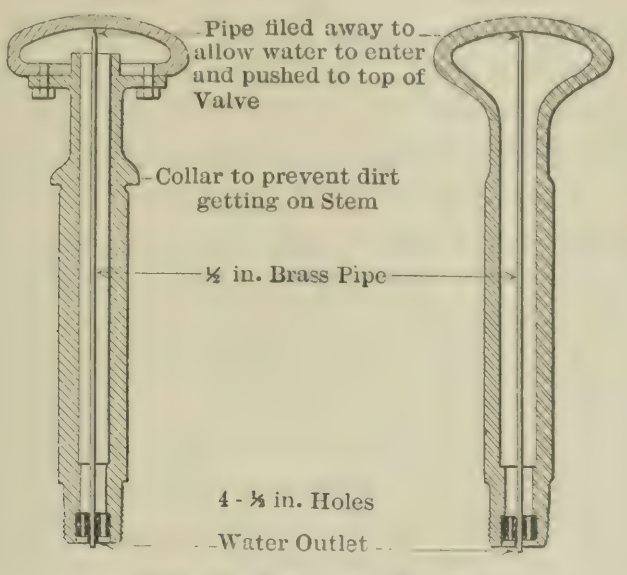

Pipe filed away to allow water to enter and pushed to top of Valve
Collar to prevent dirt getting on Stem
½ in. Brass Pipe
4 - ⅛ in. Holes
Water Outlet

FIG. 10 EXHAUST VALVES

R. H. STEVENS[1] stated that at the Carrie Furnaces they had four gas-driven blowers and five gas-electric units. After cleaning, the dust content of the gas is as low as 0.003 of a grain per cu. ft. The engines are running on about 110 to 120 cu. ft. of gas per i.h.p., the gas running from 85 to 90 b.t.u. The gas-electric engines have been in shape to operate 95 to 98 per cent of the time. The troubles have been mainly with the gears driving the lay shafts, with piston packing, rod packing, soft cast-iron cylinders, governors and inlet valves, but these have been met and overcome as they occurred.

A. N. DIEHL[2] said that the blast furnace may be considered as a producer in which some of the CO, produced near the tuyeres, is oxidized to CO_2. He described in detail the method of gas cleaning by positive and repeated spraying used at the Duquesne Works, which gives a gas with an average dust content of 0.00902. The gas after leaving the Theisen washers passes into the engines through a small gasometer. The object of this gasometer is to reduce the pressure to about 2 in. before delivering it to the mixing chambers of the engines.

[1] Mechanical Engineer Homestead Steel Works, Homestead, Pa.

[2] Superintendent of Furnaces, Duquesne Steel Works, Carnegie Steel Co., Duquesne, Pa.

He then gave a description of the engines, of which there are two power engines and four blowers, the former driving 2000-kw. generators and the latter are rated at 3600 h.p., with cylinders 42 by 60. A forced-feed lubricating system is used, operated by Richardson oil pumps. The oil is forced into the cylinder and packing cases. The packing requires a feed of one drop of oil for every other revolution of the camshaft, while the feed of the cylinder requires one drop every four or five revolutions of the camshaft. Six to eight gallons of oil are required per engine every 24 hours. A governor of the Lombard design controls the engine, the top of which actuates a small pilot valve admitting oil, under a pressure of from 175 to 200 lb. per sq. in. to a cylinder, against either one or the other end of the piston or plunger, about $3\frac{1}{2}$ in. in diameter. The governor controls the mixing valves and its operation has been very gratifying. The speed variation on the power engines is from $1\frac{1}{2}$ to 2 per cent from no load to full load, while the variation on the blowers is from 6 to 8 per cent under the same conditions.

The average thermal efficiency for the last six months of 1910 was 24.15 per cent on an average load of 1372 kw., and gave an average B.t.u. per i.h.p. of 10,529. This was only 55 per cent of the rated full load. At 84 per cent of the rated full load the thermal efficiency was 30.9, giving 8244 B.t.u. per i.h.p.

The engines are cleaned only about once every two months, and about 4 or 5 lb. of dirt scraped off the cylinder and piston heads. This dirt is a whitish deposit which seems to be held in position by carbonized oil. An analysis of it is as follows:

SiO_2	18.40	MgO	1.68
Mn_3O_4	2.40	Fe_2O_3	4.71
Al_2O_3	15.55	Ignition loss	2.17
CaO	12.20	Alkalis	42.89

The record of the past six months shows that the gas engines were required for 14,580 hr., of which they operated 14,395 hr., or 98.8 per cent, losing only 185 hr., or 1.2 per cent. Of this amount 94 hr., 40 min., is charged to the blowing tub drive, the design of which has been changed, leaving only 90 hr. and 20 min. chargeable against the engines proper, or less than $\frac{1}{2}$ per cent delay. During this time also two new engines have been put into service, the delay record covering every shutdown after the engine is first put under load. The No. 5 engine, which was started in November, operated during that month 648 hr., with only 4 hr. 20 min. delay, and this was caused by

a hot cross-head guide. The No. 6 engine, started in March, operated the balance of the month, or 526 hr. without any shutdown whatever. Of the total delays of 185 hr. before referred to, 54 were on the two power engines and 131 hr. on the three blowers.

The engines have been quite free from premature fires and backfiring, except when a variation in gas occurred, due to an irregular furnace working, and other similar occasions.

ALEX. L. HOEHR[1] said that many difficulties were encountered in starting the gas-electric plant at his works, due probably to the fact that their engines were the first of that size to be turned out by the builders. Many changes had to be made, but today and for many months past the gas engines have operated as regularly as any unit in the plant. The load factor for the seven months averaged 81.6 per cent, with a maximum of 86.06 per cent in November. There were six months when no delays were charged to the engine. The speaker proceeded to describe in detail the gas-cleaning plant in which the gas mains from the furnaces are all connected into one system, and to this system is connected the main leading to the gas-cleaning plant so that the surplus from any or all of the furnaces can pass to the gas engines.

Satisfactory as the service in this case has been, it does not prove that the gas engine is necessarily the best type of prime mover for such locations. At the time these engines were installed, the low-pressure turbine was only faintly visible on the horizon and no question was raised as to the advisability of installing the gas engine. At the present time the conditions are very different. The turbine has been developed until it is a very efficient and reliable prime mover, and the engineer cannot ignore it in figuring on new installations.

Without being able to present figures at this time to support his position, the writer believed that in the Pittsburgh district under existing conditions as to prices for coal, turbines, gas engines and labor, a complete steam plant from gas-fired boilers to low-pressure turbines can be installed and operated to produce a given amount of power for less money than would be required through a gas-engine plant. This would be due to lower first cost, lower attendance charges and lower repair charges on the steam plant. It is assumed that proper interest and depreciation charges are made to both plants.

H. J. FREYN advocated the large gas engine, calling attention to

[1] Steam and Hydraulic Engineer, National Tube Works, McKeesport, Pa.

the fact that European constructors have had the same troubles that the American gas engine users appear to have now, but have overcome them. Abroad, cylinders are used 48 in. in diameter by 55 in. stroke, these engines running from 80 to 90 r.p.m.; and he had himself seen a twin-tandem double-acting 4-cycle engine of 4000 h.p. in operation with blast-furnace gas. Rods can be designed strong enough to withstand the strains to which they are subjected. The diameter of the rods in Germany is even smaller as compared with that of the piston than in this country, and he found nickel-steel rods there containing as much as 5 per cent nickel. No cast-steel cylinders are used abroad, even in the 48-in. diameter size, and there are no troubles with cylinders cracking. The reason so far as he could judge is that foundry practice over there is a little bit different and better than here, including the selection of the iron and the method of casting. It is held over there that while the modulus of elasticity of steel castings is much higher than of cast iron, the coefficient of elongation by temperature is not much higher, so that the product of these two values, which Professor Langer calls the "Material-Ziffer," is about four times as high for cast steel as for cast iron, whereas the strength of the former material is hardly three times that of the latter. The lower this "Material-Ziffer," however, the better the casting and the less danger of cracking of the cylinder. The same is true of cast-steel pistons. He had found 48-in. cast-iron pistons in use abroad without cracking. Mr. Freyn believed that the packing for piston rods as designed in America was superior because European packings are very complicated.

As to the cost of installation, Mr. Freyn said that while it is unquestionably higher than for the steam turbine or steam engine, the result would compare very favorably when taken in connection with the high commercial efficiency of a gas engine, especially considering that the price of coal is bound to increase as it has in the past.

For ignition he would advocate an independent storage battery for any gas installation to insure reliable operation. With such an installation there is no possibility of the ignition system of the whole station failing. In connection with the question of efficiency he had found that the subject of utilization of waste heat from gas engines had made much headway in Europe. If the heat from the exhaust and the cooling water could be recovered, as in fact is now being done, the amount of energy saved would be considerable. Tests on a plant in Belgium showed that 13 per cent could be obtained from the waste heat in the exhaust and cooling water, generating low-

pressure steam with this waste heat and using it in low-pressure turbines.

E. FRIEDLANDER[1] in comparing the different prime movers from the operator's point of view, said that the present largest gas power unit of about 3000 k.v.a. capacity is not large enough for power stations of any magnitude. The very large floor space taken up by these gas engines is also a serious objection when the cost of real estate forces economy in the size of the buildings.

The overload capacity of prime movers in electric central stations, especially of turbines, is always taken into consideration in figuring on the station capacity, and plays an important part in reducing the cost of the current generated on account of the small investment necessary for steam stations to deliver a given maximum load, whereas an internal-combustion engine may be looked upon as not possessing any overload capacity whatsoever. Again it has been observed that gas engines are very slow in taking their proportionate amount of load under fluctuating load conditions, and always lag behind the other prime movers, especially turbines. This is partly caused by the extra time required to take in the charge, compress and explode it. A powerful relay governor should be arranged to reduce this lag as much as possible.

W. TRINKS, in a contributed discussion, advocated higher piston and rotative speeds, and denied the prohibitive decrease of economy that would result, insisting at the same time on the necessity of simplifying the engines.

JOS. MORGAN called attention to the importance of the load factor for the design of a plant, and suggested even the advisability of establishing a sort of electrical clearing house to supply users of current with current produced as a by-product by blast-furnace and coke-oven plants.

[1] Superintendent Electrical Department, Edgar Thomson Steel Works, Carnegie Steel Co., Braddock, Pa.

No. 1315

STRESSES IN TUBES

AN INVESTIGATION SHOWING THAT THE STRESSES IN THE WALL OF A TUBE EXPOSED TO AN EXTERNAL FLUID PRESSURE ARE OF THE SAME CHARACTER AS THOSE IN A COLUMN HAVING FIXED ENDS

By Prof. Reid T. Stewart, Pittsburgh, Pa.
Member of the Society

While engaged in planning a series of collapsing tests on commercial lap-welded steel tubing, the principal results of which are recorded in the Transactions of the Society,[1] the writer made a theoretical investigation of the stresses in the wall of a tube exposed to external fluid pressure. It was thought that the results of this theoretical investigation would aid in conducting the experiments on a more scientific basis and also serve to simplify the working up of results. The writer was led to believe that an annulus near the middle of a long tube exposed to external fluid pressure is subjected to the same kind of stress that exists in a column whose ends are fixed in direction when loaded axially. The following is a brief synopsis of this investigation, together with a comparison of the results obtained by the use of these new column formulae with the results of actual tests of columns and struts having ends fixed in direction.

2 *Apparent Theoretical Stresses in the Wall of a Tube Exposed to External Fluid Pressure.* Fig. 1 represents an annulus, 1 in. long, located near the middle of a long tube that is perfectly circular in cross-section. Let p represent the external fluid pressure in lb. per sq. in. and T the resulting tangential stress in the wall due to this fluid pressure. Now if δa represents, in angular measure, an increment of the circumference of this annulus, then an increment of the area exposed to the external fluid pressure will be $\frac{1}{2}\, d\delta a$. The

[1] Collapsing Pressures of Bessemer Steel Lap-Welded Tubes, Three to Ten Inches in Diameter, vol. 27, pp. 730–822.

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

normal pressure on this increment of area will be $\frac{1}{2}\,pd\delta a$, and the component of this pressure parallel to the line of action of the tangential stress T will be $\frac{1}{2}\,pd \sin a\delta a$. Therefore the tangential stress

$$T = \tfrac{1}{2}\int_0^{\frac{1}{2}\pi} pd \sin a\delta a = \tfrac{1}{2}\,pd \quad \ldots\ldots\ldots\ldots [1]$$

Since the tube is assumed to be perfectly circular in cross-section and of uniform thickness, this formula shows: (*a*) that the circumferential stress in all parts of any annulus of a tube exposed to an external fluid pressure is constant; and (*b*) that this constant circumferential stress per inch length of tube equals the fluid pressure in lb. per sq. in. multiplied by one-half the outside diameter in inches.

3 *Apparent Stresses in a Tube Annulus Compared with those in a Column or Strut.* Fig. 2 shows one-half of a tube annulus with the

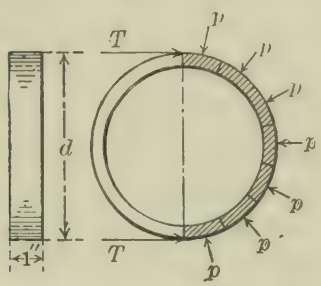


FIG. 1 ANNULUS OF UNIT LENGTH AT THE MIDDLE OF A LONG CIRCULAR TUBE

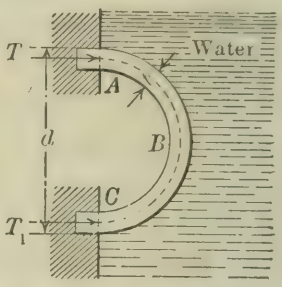


FIG. 2 HALF ANNULUS WITH FIXED ENDS

ends fixed in direction, the outside surface of the half annulus being exposed to a fluid pressure. Evidently the laws above deduced for the complete annulus apply without modification to the half annulus when its ends are fixed in direction, at the same time being free either to recede or to approach each other. All portions of this half annulus, then, are subjected to the compressive stress T (Formula 1) which acts circumferentially in the direction ABC. Opposing this, of course, is the equal circumferential stress T_1, acting in the direction CBA. By straightening the half annulus (Fig. 2) so as to bring A, B and C into the same straight line, the column or strut shown in Fig. 3 will result. It is evident that the forces T and T_1 are each rotated by this action through 90 deg. This shows that the theoretical stresses in a tubular half annulus are identical with those of the column having ends fixed in direction, when the length of the column equals the

mean semicircumference of the annulus, the two, of course, having the same cross-section.

4 It will become evident from a comparison of Figs. 4 and 5 that we should take for the length of a column having fixed ends, the half circumference of an annulus located near the middle of a long tube exposed to an external fluid pressure. Fig. 4 shows the ideal collapse section for all the long tubes tested. It will be observed that the tangents to the annulus at both the highest and lowest points before collapse, as shown by the dotted lines, are precisely parallel to the tangents at the same points after collapse. In other words, these portions of the annulus while undergoing deformation remain fixed in direction. Comparing this with Fig. 5, which shows the most

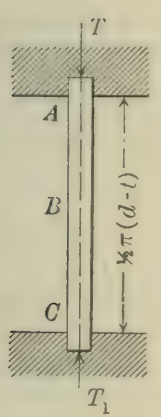


Fig. 3

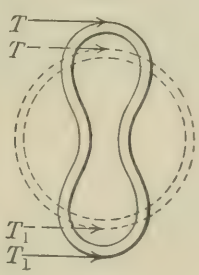


Fig. 4

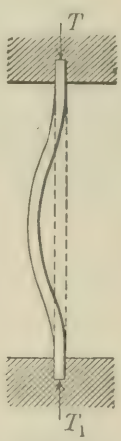


Fig. 5

Fig. 3 Strut representing Half Annulus Straightened
Fig. 4 Ideal Collapse Section for Long Tubes
Fig. 5 Most Probable Manner of Column Failure

probable manner of buckling of a long column having fixed ends, it will be seen that the two are identical as regards apparent stresses for the conditions above stated, namely, when the annulus is perfectly circular and of uniform cross-section, the length of the column with fixed ends being equal to the mean half circumference of the annulus.

5 As the tube annulus departs from the circular form, while failing under fluid collapsing pressure, new stresses arise which have no counterpart in the equivalent column. An investigation has shown that these stresses are slight for small departures from roundness, so that for commercial tubing exposed to external fluid pressure the stresses are substantially the same as those in an equivalent column as illustrated above.

6 *The Author's Formula* [*B*] *for the Collapsing Pressures of Steel Tubes Reduced to an Equivalent Column Formula.* In order that he might be able to test the accuracy of the above theoretical deductions, the author has transformed his formula [*B*] for the collapsing pressures of lap-welded Bessemer steel tubes[1] so that it may be used for calculating the crippling strength of columns or struts with fixed ends. This transformation was effected as follows:

7 Referring to Figs. 2 and 3, which represent respectively a half annulus of the tube and its equivalent column or strut, it is evident that the length of the equivalent column will be

$$l = \frac{\pi}{2}(d-t) \text{ or, } \frac{d}{t} = \frac{2l}{\pi t} + 1 \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

where d and t represent respectively the outside diameter and the thickness of the wall of the tube in inches and π the ratio of the circumference to the diameter of a circle. Since $t = 3.464r$, where r equals the radius of gyration of cross-section of the annulus and of its equivalent columns

$$\frac{d}{t} = \frac{2l}{3.464\,\pi r} + 1 = 0.1838\,\frac{l}{r} + 1 \quad \ldots\ldots\ldots\ldots [3]$$

$$p = \frac{2T}{d} = \frac{2tS}{d} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

where p represents the external fluid pressure in lb. per sq. in., T the total circumferential stress in the wall of the annulus 1 in. long, and S the total circumferential stress per sq. in. of cross-section, both being expressed in lb. Also, t and d, as before, represent respectively the thickness of the wall and the outside diameter of the tube, in inches. Formula [*B*], using the same notation as before, is

$$p = 86{,}670\,\frac{t}{d} - 1386 \quad \ldots\ldots\ldots\ldots\ldots\ldots [B]$$

By equating the second members of equations [4] and [*B*] we get

$$\frac{2tS}{d} = 86{,}670\,\frac{t}{d} - 1386$$

from which

$$S = 43{,}335 - 693\,\frac{d}{t} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

[1] Trans.Am.Soc.M.E., vol. 27, p. 793.

By substituting the value of $\frac{d}{t}$ from equation [3] we get

$$S = 42{,}642 - 127.4\frac{l}{r} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [K]$$

8 This is a formula for the crippling strength of a column with fixed ends, as derived directly from formula [*B*] for the collapsing pressures of long tubes that are exposed to external fluid pressures. In this formula, S represents the axial load on the column in lb. per sq. in. of cross-section, while $\frac{l}{r}$ represents the slenderness ratio, or the length of column divided by the least radius of gyration of cross-section, both being expressed in the same linear unit.

9 Since formula [*B*] is applicable to values of thickness divided by outside diameter $\left(\frac{t}{d}\right)$ greater than 0.023, formula [*K*] should be applicable to values of length of columns divided by least radius of gyration, $\frac{l}{r}$, less than

$$\frac{\frac{\pi}{2}\frac{(d-t)}{t}}{3.464} = 1.732\ \pi\left(\frac{d}{t} - 1\right) = 1.732\ \pi\left(\frac{1}{0.023} - 1\right) = 230$$

Note that formula [*G*][1] is tangent to formula [*B*] at $\frac{t}{d} = 0.024$, which gives a slenderness ratio $\left(\frac{l}{r}\right)$ at point of tangency of 221, which latter should therefore be the true limiting value of $\frac{l}{r}$ for formula [*K*] when used in connection with formula [*L*] as given below.

10 *The Author's Formula* [*G*] *for the Collapsing Pressures of Steel Tubes Reduced to an Equivalent Column Formula.* In a manner similar to the above derivation of formula [*K*] the author's formula [*G*] has been transformed into an equivalent formula for the crippling strength of long columns or struts. Using the same notation as before collapse formula [*G*] which is applicable[1] to values of $\frac{t}{d}$ less than 0.023, is

$$p = 50{,}210{,}000\left(\frac{t}{d}\right)^3 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [G]$$

[1] Trans.Am.Soc.M.E., vol. 27, p. 795.

and its equivalent column formula, derived in a similar manner, is

$$S = 25{,}105{,}000 \left(\frac{1}{0.1838 \frac{l}{r} + 1} \right)^3 \quad \ldots\ldots\ldots\ldots [6]$$

which is applicable to values of $\frac{l}{r}$ greater than 221, as stated above. This somewhat complex formula is represented with sufficient accuracy for all practical purposes by the simple formula

$$S = \frac{708{,}000{,}000}{\left(\frac{l}{r}\right)^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [L]$$

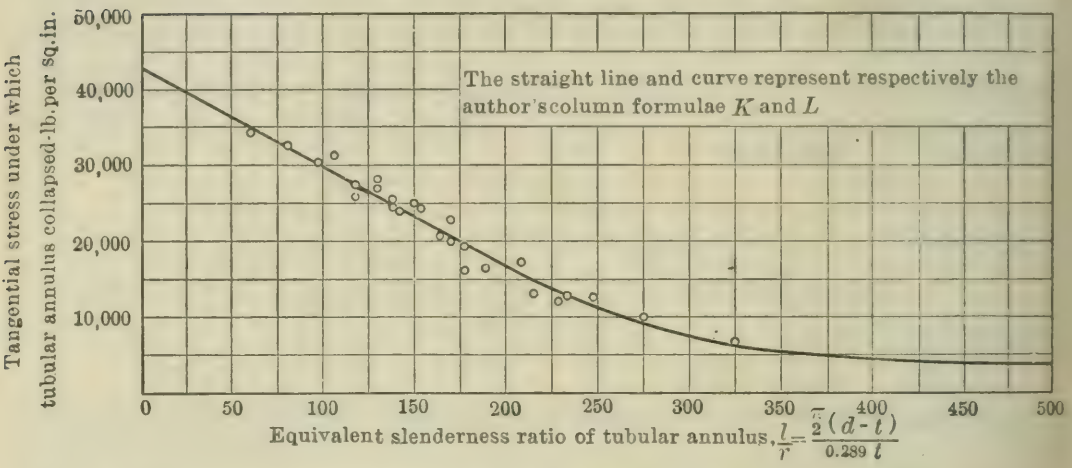


FIG. 6 PLOTTED RESULTS OF THE AUTHOR'S EXPERIMENTS ON COLLAPSING PRESSURES OF LAP-WELDED STEEL TUBES

where S as before, represents the axial load on the column in lb. per sq. in. of cross-section, and $\frac{l}{r}$ the length divided by least radius of gyration of the column, both being expressed in the same linear unit. This formula applies only to columns having both ends fixed in direction and for values of $\frac{l}{r}$ greater than 221.

11 *Verification of the Author's Column Formulae* [*K*] *and* [*L*] *by Comparison with Results of Tests on Columns.* In order to show that the new column formulae given in this paper are applicable to commercial shapes and annular sections when used as columns with ends xed in direction, Figs. 6 and 7 were prepared. The only tests on

commercial struts and columns with fixed ends known to the writer are those made by James Christie on wrought-iron struts,[1] and those made at the Watertown Arsenal[2] in 1909.

12 The average physical properties of the iron in the struts tested by Mr. Christie were

Tensile strength, lb. per sq. in.	49,000
Elastic limit, lb. per sq. in.	32,000
Elongation in 8 in., per cent	18

while those of the steel constituting the lap-welded tubes tested by the writer and at the Watertown Arsenal were

Tensile strength, lb. per sq. in.	58,000
Yield point, lb. per sq. in.	37,000
Elongation in 8 in., per cent	22

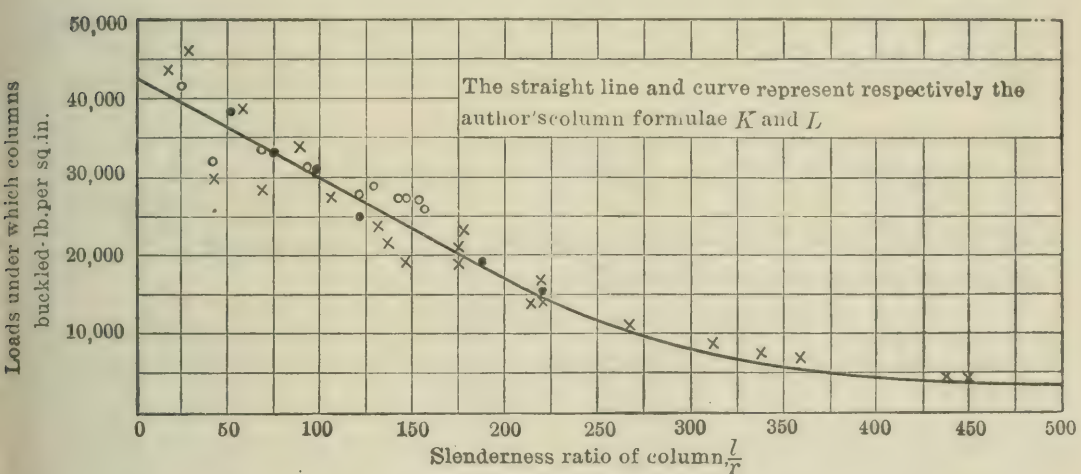


FIG. 7 PLOTTED RESULTS OF VARIOUS COLUMN EXPERIMENTS

(x) indicates plotted results of Christie's experiments on wrought-iron angles ranging from 4 in. × 4 in. × ⅜ in. to 1 in. × 1 in. × ⅛ in., with fixed ends.

(•) indicates plotted results of Christie's experiments on lap-welded wrought-iron tubes 2.87 and 2.37 in. outside diameter used as columns with fixed (flanged) ends.

(o) indicates plotted results of Watertown Arsenal experiments on lap-welded steel tubes, 5 in. outside diameter, used as columns with fixed ends.

13 These data show that the material of the angles and tubes tested by Mr. Christie as compared with those of the tubes tested by the writer and at the Watertown Arsenal had average physical properties less by 15 per cent in tensile strength, 13 per cent in elastic limit, and probably less than 10 per cent in modulus of elasticity, or rigidity factor. It should be remembered while comparing the results

[1] Trans.Am.Soc.C.E., 1884, p. 117.

[2] Proceedings, American Society for Testing Materials, 1909, p. 413.

of these experiments, that these differences in the physical properties of the materials, which it will be noticed are comparatively small, will be more or less offset for two reasons: (*a*) the tubular annulus at the point of failure varies somewhat more from being truly circular than does the strut from being truly straight; and (*b*) there is a small bending moment on the wall of the tubular annulus directly due to the action of the external fluid pressure on an annulus that is slightly out of round, for which there is no counterpart in the equivalent column.

14 Fig. 6 represents the plotted values of the results of the writer's experiments on the collapsing pressures of lap-welded steel tubes. These results are plotted to a horizontal scale representing the equivalent slenderness ratio of the semi-tubular annulus considered as being under the same conditions of stress as a column with ends fixed in direction (Figs. 2, 3 and 4). Since the mean semicircumference of the tube annulus equals $\frac{\pi}{2}(d-t)$ and the radius of gyration of the section equals $0.289t$, this slenderness ratio will be $\frac{l}{r} = \frac{\frac{\pi}{2}(d-t)}{0.289\,t}$. The vertical scale represents the apparent tangential stress T (Fig. 2), under which the tubular annulus actually failed.

15 Fig. 6 represents the plotted values of the group averages of Series 2 of the author's experiments on the collapsing pressures of lap-welded steel tubes, 3 to $12\frac{3}{4}$ in. outside diameter.[1] The straight portion of the line represents the writer's column formula [*K*] plotted to the same scales, while the curved portion similarly represents column formula [*L*].

16 Fig. 7 represents the plotted values of all results of tests on commercial struts and columns with ends fixed in direction known to the writer. It should be noted here that practically all tests of commercial columns and struts have been made with flat, pin, or round ends; and therefore do not represent the conditions involved in this comparison.

[1] Trans. Am.Soc.M.E., vol. 27, pp. 787-802.

No. 1316

THE PURCHASE OF COAL

By Dwight T. Randall, Boston, Mass.

Member of the Society

Most boiler rooms are now conducted in a manner which permits of considerable saving along two lines: (*a*) the selection of a coal which is suited to the plant and at the same time is capable of delivering the greatest amount of heat to the boiler for a unit of cost; (*b*) burning the coal by approved methods to obtain the highest practical efficiency.

2 The coals which are offered in almost any market vary in price and in quality to an extent which justifies a careful study of their character and heating value in order to determine which coal will prove most economical when the equipment, the load conditions and the price are considered. A coal which is entirely satisfactory in one plant may be unsuited to another. It is possible to burn almost any fuel with reasonably good efficiency provided the furnace is properly designed for the particular fuel to be burned.

3 Coals which are suitable for any given equipment depend for their value principally upon the B.t.u. and the size of the coal. The B.t.u. or heating value of coals of any given type determine directly their value as fuels. When coals of the same character are under consideration the heating value may be considered as a correct measure of the value of the coal. When coals of different characters are to be compared, the character of the coal as well as heating value must be considered.

RELATION OF QUALITY TO RESULTS

4 The influence of the volatile matter on the efficiency depends on the design of the furnace. With a poor furnace and indifferent firing the coals containing about 18 per cent volatile matter may give re-

In abstract form. Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers. The complete paper appears in The Journal, March 1911, and the discussion, September 1911.

sults 10 or 12 per cent higher than coals containing 30 per cent or more volatile matter. With furnaces adapted to the kind of coal burned there is but little loss of combustible gas.

5 The ash in the coal affects the heating value to some extent as there is a loss of both time and heat while the fires are being cleaned and the presence of large quantities of ash interferes with the proper distribution of air through the fuel and may lower the efficiency.

6 The moisture not only requires heat to evaporate it into steam, but if the coal is very wet and is fired in large quantities, it may cool the bed of fire and cause an additional loss of unburned gas.

7 The size of coal is important in many cases. If the coal does not coke and is fine, there may be a large loss of fuel through the grates when burned on inclined grate stokers or on hand-fired grates at rates that require frequent breaking up of the fuel bed. The size of the coal also affects the economy with which it may be fired. If coal is too large more air is admitted than is necessary to burn it properly and if the fuel bed can not be increased in thickness to overcome this difficulty, there will be a large heat loss. If the coal is fine and the draft is very strong, some of it will be carried off the grate only partially burned. This is frequently the case when the coals are fine and light and the boilers are forced.

8 Fine coal which cakes and forms a porous coke may be burned with good efficiency. If the coal does not coke but packs closely on the fuel bed, it is difficult, if not impossible, to secure a uniform air supply at all parts of the bed and the combustion is poor owing to an excess of air at some points and a lack of air at others.

9 Fuels considered without reference to any particular equipment may be valued on the basis of their available heating value. It has been found possible to design furnaces to burn almost any fuel with reasonably good efficiency when based upon the available heat of the fuel. This has been accomplished with tan bark, sawdust, lignite and low grade coals. As a rule inferior coals can be bought much more cheaply on their heating value than the higher grades of coal and it is to the interest of every consumer to select the coal which will give the greatest amount of heat from a unit cost, provided it can be burned successfully in his plant. In many cases it will be profitable to change the equipment so as to burn slack coal or coals which are below the average quality. It is fully as important to take into account the size and character of coal when automatic stokers are in use as when the coal is hand-fired.

10 The same intelligence should be used in selecting coal for a

given use as is required in selecting steels for manufacturing purposes. Some of the most progressive users have been aware that there was a considerable difference in the quality of the coals used in their plants and have generally decided that the chemical analysis, with determinations for heating value, is the best basis on which to establish a standard.

11 The methods employed in burning the coal are of equal importance with the quality and should be given careful attention. The coal dealer should not be held responsible for results in boiler plants except as influenced by changes in the quality of the coal delivered. A coal which is suited to one plant may not burn well in another, owing to differences in equipment, load condition or to the methods of handling the fires.

12 The advantages of knowing accurately the quality of coal which is being burned in a power plant and whether any changes in the coal consumption are due to the coal or to the method of operating the plant, have led a number of managers to make analyses of all the coal delivered to their plants. After following the deliveries in this way for a year or more, these men have as a rule decided to place their contracts with the understanding that if the coal delivered can be prepared so as to eliminate more of the impurities, they will pay a higher price, and if the coal is below quality they will deduct from the regular price in accordance with the quality of the coal.

13 Immediately on considering the purchase of coal on a guaranteed analysis the question as to how the sample shall be taken and by whom it shall be analyzed is raised. The method of taking a sample of coal is fully as important as the manner in which it shall be analyzed and the cause of doubt as to the value of coal analyses has been largely due to ignorance or carelessness in taking samples for analysis. It is only fair to both parties concerned that the sample should be taken in the manner which will secure a small portion which is thoroughly representative of the entire lot. A method has been quite generally adopted and experience has shown that when two samples are taken in accordance with these approved methods the results are within reasonable limits of accuracy.

METHODS OF SAMPLING

14 The following method of obtaining a sample of coal has been used by a number of different firms and has been found satisfactory. The main object in taking a sample of coal is to secure a small por-

tion of the coal which represents as nearly as possible the entire shipment or delivery.

15 The original sample should preferably be collected in a large receptacle with cover attached, by taking small shovelfuls from many parts of the car, barge or vessel as it is being unloaded, or from as nearly all parts of a pile as possible, care being taken in all cases to secure practically the same amounts from the top, middle and bottom of the coal. The original sample thus taken should amount to 500 lb. or more, preferably 1000 lb. to 2000 lb. A separate sample should be taken from each 1000 tons or less delivered. The gross sample thus collected should contain the same proportion of lump and fine coal as exists in the whole shipment. It should be protected from the weather in order to avoid gain or loss in moisture and should be immediately quartered down to a smaller sample, according to the following method:

16 The large lumps of coal and impurities should be broken down on a clean, hard, dry floor with a suitable maul or sledge. The coal should be thoroughly mixed by shoveling it over and over and formed in a conical pile. The pile should then be quartered, using a shovel or board to separate the four quarters. Two opposite quarters should then be rejected and the remaining two broken down to a smaller size, mixed and re-formed in a conical pile and quartered as before. This process should be continued until the lumps are $\frac{1}{4}$ in. in size or smaller and a one or two-quart final sample remains. *All* of this final sample should immediately be placed in one or more glass or metal cans and sealed air tight. The following table gives the largest sizes allowable in the samples of various weights and the coal should preferably be broken into still smaller sizes before quartering:

Weight of Sample	Should Pass Through
1000 lb. or over	$1\frac{1}{2}$-in. sieve
500 lb. or over	$1\frac{1}{4}$-in. sieve
250 lb. or over	1 -in. sieve
125 lb. or over	$\frac{3}{4}$-in. sieve
60 lb. or over	$\frac{1}{2}$-in. sieve
10 lb. or over	$\frac{1}{4}$-in. sieve

17 The sample should be worked down as rapidly as possible to avoid loss of moisture through exposure to the air. The outside of the can should be plainly marked and a corresponding description placed inside the can.

18 The following data should accompany the sample:

Coal delivered to.....
Sampled by.....Date.....
Amount taken for original sample.....
Amount of coal sample represents.....
Sampled from barge, car or pile.....
Car (initial and number).....
Barge or vessel.....
Trade name of coal.....
Grade (¾, slack, nut, run of mine, etc.).....
Remarks (appearance of coal, lumps, slate, sulphur balls, weather conditions, etc.).....
Sold by.....Mined by.....
County.....State.....Mine.....

19 It is difficult to make the average man understand the importance of the sample and the influence of the method of sampling on the results. The methods and care used in breaking, down and quartering the sample are very important. By the use of careful methods, however, it is generally agreed that a sample of coal, taken by an approved method and analyzed by an experienced coal chemist, should show results which, when compared with the true values, are within the following limits:

Moisture.....1.00 per cent of the coal as delivered
Ash + or −.....0.50 per cent of the dry coal
Sulphur + or −.....0.10 per cent of the dry coal
B.t.u. + or −.....1.00 per cent of the dry coal

20 When samples are taken in a proper manner, the results will be sufficiently accurate for all commercial purposes and are within the limits which are found in the comparatively simple operation of determining the weights of the coal shipped. It is hardly worth while to adopt more costly methods of sampling coal in order to secure a greater accuracy, until the methods of weighing coal are improved and the accuracy of the weights guaranteed within less than 1 per cent.

THE SELECTION OF COAL

21 The problems of purchasing a supply of fuel for any given plant, so as to obtain a coal that is suitable for the equipment in use and one that will deliver the greatest amount of heat to the boiler

for each dollar expended, is one which requires experience and an intimate knowledge of various kinds of equipment for burning coal, and also of the different characteristics of the coals available at reasonable freight rates.

22 The following information should be considered by the engineer in deciding on the best coal for a plant:

a Kind and size of boilers and furnaces
b Load conditions, average and maximum loads
c Draft available and how controlled
d Character of the coals offered or available
    (1) Moisture and its effect on weight of combustible matter delivered
    (2) Volatile matter and its relation to kind of furnace
    (3) Ash: its amount and its fusibility and tendency to clinker
    (4) Sulphur: the amount and how combined
    (5) Heating value in B.t.u.
    (6) Coking qualities of the coal
e Size of the coal
    (1) Relation of the size of coal to the equipment

SPECIFICATIONS FOR COAL

23 After it has been decided what kind of coals may be burned successfully in any given plant, it is important that the specifications be so drawn that it will be to the interest of the dealer to deliver the kind of coal which has been established as standard in his proposal and prevent the substitution of lower grades of coal which might be difficult to burn with good results. It is evident that a specification based on heating value alone will not do this and that there should be some clause making it possible to reject the coal, or to burn it and pay for it at a reduction in price greater than that due to B.t.u. only.

24 There has been a great deal said for and against the plan of purchasing coal on a guaranteed analysis and, as is often the case, both sides are right but they are really discussing different things.

25 A properly drawn specification protects the dealer who is prepared to furnish good coal in competition with dealers handling inferior coals at the same price. Where these specifications permit the coal contractor to state the analysis of his coal which on acceptance of the bid becomes the standard for the contract, there need be

but little variation in the price if the dealer is familiar with the analysis of the coal offered, and if the standard is based on average values the premiums and penalties for the year should practically balance each other. Many dealers have bid on impossible analyses and then blamed the specification plan for their losses.

26 A properly drawn specification providing for premiums for better coal than that specified, encourages the coal operators to exercise greater care in mining and in picking the coal before shipping, and enables them to secure a return on the cost of such preparation. Most consumers have found that it is not profitable to pay freight on an unnecessary amount of slate and ash in the coal.

27 The important items in a specification are as follows:

a A statement of the amount and character of the coal desired.

b A statement regarding the conditions for delivery of coal.

c A statement regarding the disposition which will be made of the coal in case it is outside the limits specified.

d A statement regarding the corrections in price for variations in heating value, for variations in ash and for variations in sulphur, provided it is found advisable to limit the percentages of ash and sulphur in the coal to be delivered.

e A blank form on which the dealer may submit price and the kind and quality of coal which he proposes to furnish.

28 There are several forms which have been prepared along these lines which have proved satisfactory. It is necessary in almost every case to modify the specifications to fit the special conditions in the plant and the fuels which are available.

29 In the past many dealers not familiar with the quality of the coals have bid on contracts and guaranteed a quality of coal that was better than can be delivered from any mine in the United States. Naturally these analyses that had been useful as exhibits were found to be poor standards on which to base the guarantees of coal to be delivered. Many progressive dealers have recognized the reasonableness of the demand for a standard for quality of the coal to be delivered and they are selling coal on a basis which secures for them the average price they expect to get for the coal.

30 The importance of testing coal purchased under contract may be illustrated by two recent cases. In Case 1 the coal was guaranteed to be Georges Creek and in Case 2 to be New River (Table 1).

31 Neither of the parties in the above cases had made a practice of having the coal which was delivered at their plants, sampled and analyzed. In such cases the blame for paying a good price for a poor

coal rests with the purchaser. The dealer probably knew as little about the quality as the man who paid for it.

32 The plan of purchasing coal on a specification based on a guaranteed analysis may or may not be a good one, depending on circumstances. It has proved satisfactory in nearly every case when the specifications were so drawn as to protect the buyer against substitution of other coals and at the same time was perfectly fair to both the buyer and to the seller. Such specifications actually protect the

TABLE 1 DATA OF COAL PURCHASED UNDER CONTRACT

	Case 1		Case 2	
	Guaranteed Analysis as Delivered	Coal Delivered	Guaranteed Analysis as Delivered	Coal Delivered
Ash in dry coal	8.00	12.66	6.00	8.48
B.t.u. in dry coal	14,250	13,558	14,700	13,981

dealer against unfair competition as has been shown in many cases. This plan will not be satisfactory if the specification is carelessly drawn and the coal is selected on the basis of price without regard to its adaptability to the furnaces; nor will it prove satisfactory if the sampling is done by ignorant or careless men and the analysis made in poorly equipped laboratories by inexperienced chemists. Whether it will be advisable to purchase for a plant on this basis depends on the amount of coal used, the amount delivered at one time, the kinds of coal available and whether the purchaser and the dealers are qualified by a knowledge of the available coals to enter into a contract on this plan.

DISCUSSION

C. W. Rice, speaking as a member of the Committee of the Society on the Conservation of Natural Resources, said he desired to emphasize, without in any way conflicting with the idea of this paper, the importance of designing plants to use low-grade fuels. The tendency of the paper is to direct purchasers to be particular with the coal dealer, with the result that in the effort to meet specifications only 14,500 B.t.u. coal is furnished, and there is no market for the low-grade material. Hand in hand with the idea of purchasing coal on the heat unit basis should go the designing of the plants to fit market conditions, taking advantage of the fact that the coals of slightly less B.t.u. than the best are materially less expensive.

Engineers dealing with the forces of nature have a duty to direct their work along the lines of conservation, an idea inclusive of the efficiency of those forces.

CHARLES WHITING BAKER emphasized the points brought out by Mr. Rice and called attention to the fact that in the East a number of plants are successfully burning a very low grade of fuel, which it would be impossible to burn by ordinary methods, through the use of the steam jet blower. While the use of the steam jet may be criticized from a thermodynamic point of view, its advantage lies in the fact that it keeps the fuel bed cool enough to prevent trouble with clinkering.

R. C. CARPENTER held that the ideas just expressed did not constitute an argument against the necessity for testing coal or for purchasing coal by analysis. He had lived for a number of years in the district bordering the anthracite coal region, where it was necessary to burn the small grades of anthracite that were in little demand for general purposes, and he was very familiar with the methods employed in burning this low-grade fuel. The material contains large quantities of slate for which thousands of dollars are spent annually, and the loss in this respect can be stopped only by the purchase of coal on analysis. He believed the paper to be just as valuable from the standpoint of the utilization of poor fuel as from that of the purchase of the more valuable kinds.

W. F. M. Goss agreed with this point of view, and believed that the coal producer must take more responsibility for the suitable preparation of his coal before delivering it to the consumer. The poor coal should, of course, be mined and used, thereby carrying out the policy of conservation, but before the delivery of this material, the operator should be encouraged by the development of markets which will take a superior fuel, to improve its quality by washing, sorting, and even by briquetting.

E. D. MEIER stated that he was a firm believer in the analysis of coal, and the practice of basing the purchase price on the analysis, provided, however, that this policy be applied only to a particular district. Analyses cannot justly be applied to comparison of coals from different districts because other things besides the chemical composition of the coal must be considered. For instance, a certain coal in Illinois contained as an impurity a bituminous shale which

carried 75 per cent ash, and although this coal may have compared favorably with other coals on a B.t.u. basis, this was not a true comparison, because the shale fused and choked the fire, making it necessary to clean the grates very frequently. Again, coal from a certain Indiana mine high in B.t.u. gave poorer results in boiler furnaces than coal from other districts lower in B.t.u. because it was extremely friable, causing much loss by dropping through the grates.

E. W. RUTHERFORD wrote that two contracts for coal purchased on specifications had recently come to his attention which he thought might be of interest in this connection. Two different dealers guaranteed the same B.t.u. and furnished coal of the same trade name. While these contracts have not terminated, practically all of the coal due has been delivered. The analyses show that one dealer will receive a bonus of from $0.01 to $0.02 per ton and that the other dealer will be penalized about $0.10 per ton, the two contracts showing a difference in value of $0.11 or $0.12 per ton, or about 3 per cent. Since approximately 20 samples were taken in each case by experienced men, there should be no appreciable difference due to sampling.

Undoubtedly, in some cases more than 3 per cent difference exists in the full value of coals sold under the same trade name when purchased on the heat unit basis. This leads to the question how much more, if any, it is necessary to pay for coal purchased on the heat unit basis than for coal purchased in the ordinary way. If the plant is favorably located the cost per ton should not be increased, because the specifications should protect the dealer as well as the purchaser, as pointed out by Mr. Randall in Par. 25. The "as received" basis is occasionally objected to by the dealer and does not seem fair in all cases, as, for example, in all-rail shipments, but is undoubtedly preferable to the dry basis when the coal can readily be sampled at the loading and weighing point.

If the author can give us any further information on these points, it will be appreciated.

PERRY BARKER, representing Mr. Randall, said in reply to Mr. Rice's criticism of the paper, that if the plant can be so designed and the fuel is so available that a low grade can be used, it will be profitable to change the equipment to suit low-grade fuels. However, the variation in this coal is such that it is not profitable for the average plant to attempt to burn it without special attention.

He cited as an example the small size of anthracite which comes to the New England market, ranging all the way from 14 to 24 per cent of ash. It would be conserving some of our natural resources if that 24 per cent of ash could be burned at all efficiently. If the ash be kept down to 18 per cent, probably the coal is a good proposition in a plant, either burned alone or mixed with a certain portion of good grade of bituminous coal, but if it comes to the market with from 14 to 25 per cent of ash there should surely be some correction for this variation in quality.

In reply to Mr. Baker's suggestion that the steam jet be used in burning low-grade fuel, Mr. Barker said that if it is in the hands of an experienced operator, it is a very efficient piece of apparatus, by the use of which a small size of anthracite may be burned. He had found a number of cases where the steam jet was a very dangerous thing in a power plant on account of the improper regulation.

Professor Carpenter had touched on the question of the variation of ash in small sizes of hard coal which are marketed. It is reasonable to suppose that some penalty should be inflicted if a large percentage of ash is going to be delivered in coal.

Professor Goss had spoken in regard to the improvement of the product along the lines of preparation. The seller who will take care to pick or wash his coal properly will receive compensation accordingly. A mine in Pennsylvania ran about $1\frac{1}{2}$ to $2\frac{1}{2}$ per cent higher in ash than an adjacent mine, due to a change in the nature of the seam. The quality of the miners' work compared favorably with their neighbors', and in order to reduce their ash, a picking table was put in and improved methods of mining were introduced, so that they reduced their ash 2 to 3 per cent, which put them on the same basis as their neighbors and on whatever coal they were selling on specifications they were obtaining compensation accordingly.

President Meier's remarks in regard to the analyses of fuels are along the line of notations in the paper in regard to the character of the coal, which may show the same heat value and the same percentage of ash and of volatile matter, but the nature of this last should be considered as well as the amount. Several coals may also contain the same percentage of ash but the relative proportions of bases and acids which compose the ash in these different coals will govern the tendency to produce fusion or clinker.

Replying to Mr. Rutherford's question in regard to the amount of the difference in quality of coals sold under the same general trade

name, the figures submitted in Table 1, together with some additional data, Table 2, may be cited as evidence of the extent of this difference.

It will be noted from Table 2, that the quality guaranteed on the specifications in each case was about equal to the average of the two coals delivered on large tonnages. Differences much larger than those submitted in the above table have been cited from time to time but

TABLE 2

	Case 1		Case 2	
	% Ash	B.t.u.	% Ash	B.t.u.
Quality guaranteed under specification cited	8.00	14250	6.00	14700
Quality delivered under specification cited	12.66	13558	8.48	13981
Average analysis of large tonnages to other consumers	8.51	14250	6.41	14543
Single delivery showing highest quality	7.48	14487	4.54	14890
Percentage difference in value between highest delivery and coal furnished under specification cited	6.5*		6.2*	

*Based on quality guaranteed

these are generally due to particularly poor preparation or inaccuracies in sampling or analysis.

In regard to the increase in price per ton when coal is purchased on a heat unit basis, a number of cases due to various causes have been reported. The principal reason for making any increase when coal is purchased by this method is due to ignorance of the dealer in regard to the quality of the coal which he will be able to deliver.

Some years ago it was the universal practice in most localities to increase the price of coal when purchased on a heating value basis but the large number of analyses of the various products which have been made in recent years have established values for most of the well known coals and dealers are willing to bid on the premium or penalty basis without increasing the price.

No. 1317

ENERGY AND PRESSURE DROP IN COMPOUND STEAM TURBINES

CALCULATIONS OF THE DISTRIBUTION OF ENERGY AND PRESSURE DROP AMONG THE STAGES OF A COMPOUND IMPULSE TURBINE

By Forrest E. Cardullo, Durham, N. H.

Member of the Society

In a paper[1] read before the meeting of the Society of Naval Architects and Marine Engineers in Detroit, June 1909, Prof. C. H. Peabody explains a method of determining the pressure in the steam chests of the several stages of a compound impulse turbine. As a substitute for this method the writer submits the following, suggested largely by the method given in the above mentioned paper, which he believes to be both more simple and more exact in application.

2 The general methods open to the designer for determining the properties of steam during its passage through the turbine are three in number. He may make use of empirical equations giving the relation between the heat content, entropy and temperature or pressure of expanding steam, as was suggested by Dr. Steinmetz.[2] He may make use of Mollier's total heat-entropy diagram which gives the relation between the total heat, entropy, pressure and quality of steam. Or he may make use of a table like that of Professor Peabody giving the relation between the temperature, entropy, total heat, quality and specific volume of steam. The last two methods are both more simple and more accurate than the first one and are to be preferred.

3 Assume that a turbine is to be designed having n stages and that the diameters of the moving elements of each stage are the same.

[1] A Method of Determining Pressures for Steam Turbines, Trans. Soc. Naval Archs. and Marine Engrs., 1909, p. 32.

[2] Proc. Am. Soc. M. E., March 1908.

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

Then, in order to have the steam issue from each set of nozzles at the same velocity, it is necessary to have the same heat drop in each stage. The heat drop per stage will, of course, be $\frac{1}{n}$ of the total heat drop. Were there no retransformation of work into heat, it would be necessary only to find from an entropy table or diagram the entropy and total heat of the steam as it enters the turbine, and the total heat, at the same entropy, of steam of the terminal pressure, to subtract the second quantity from the first in order to obtain the total heat drop, and then to divide this drop by the number of stages to obtain the heat drop per stage. The pressures in each stage would then be found by subtracting the heat drop per stage n times from the initial heat content and finding from the table or diagram the pressure of steam having the heat content so found, at the given entropy.

4 This method may be illustrated by the following problem, taken from Professor Peabody's paper: Assume that the initial steam pressure is 164.8 lb. per sq. in. and the final pressure is 1.005 lb. per sq. in., that the steam is initially dry and saturated, and that the number of stages is two. From Professor Peabody's table, we find the initial entropy to be 1.56 and the initial heat content to be 1193.3 B.t.u. The heat content of steam of 1.56 entropy at the terminal pressure is 871.1 B.t.u. The difference between the initial and final heat content, or the heat drop, is 322.2 B.t.u. The heat drop per stage is one-half this or 161.1 B.t.u. The heat content of the steam entering the second stage is $1193.3 - 161.1 = 1032.2$ B.t.u. The pressure of steam having this heat content and the entropy 1.56 is 18.4 lb., which would be the absolute pressure of the steam as it enters the steam chest of the second stage.

5 In the actual steam turbine, we find that the quantity of heat transformed into work is only 40 per cent to 70 per cent of the heat theoretically available for transformation by isentropic expansion. A small portion of this is lost by gland leakage, radiation and bearing friction. The most of the missing energy, however, has been retransformed into heat by eddying, fluid friction, blade leakage, etc., and appears in the steam, increasing its entropy. We may assume that in actual practice 60 per cent of the energy theoretically developed in the first stage of this turbine, or 96.6 B.t.u., would be transferred to to the rotating member, and about 40 per cent, or 64.5 B.t.u., would be retransformed into heat, making the heat content of the steam entering the stage, $1193.3 - 96.6 = 1096.7$ B.t.u. This would give for

the entropy of the steam at the pressure of 18.4 lb., the value 1.655. The heat content of steam of 1.655 entropy and 1.005 lb. pressure is 925.0 B.t.u., which gives for the heat drop in the second stage 1096.7 − 925.0 = 171.1 B.t.u. This is more than 6 per cent greater than the heat drop assumed for the first stage. It is plain that in order to equalize the heat drop in the two stages, the pressure range in the first stage must be increased at the expense of that in the second stage. It is also evident, as Professor Peabody has pointed out, that the efficiency of each stage is less than the efficiency of the turbine as a whole, the larger efficiency of the turbine being accounted for by the fact that the steam entering the second and each subsequent stage contains an extra quantity of heat as a result of the inefficiency of the preceding stages, which is available for transformation into work of the succeeding stages.

6 It will be found by trial and adjustment that if the theoretical heat drop per stage, which we may designate by the symbol $\frac{\Delta H}{n}$, be multiplied by the empirical factor $1 + K$, we will have the heat drop per stage which will give an actual equality in the quantity of energy developed in each stage. In the above empirical factor, the value of K is found by the equation

$$K = 0.00056 \left(\frac{n-1}{n}\right) \Delta H \, (1-E)$$

In this equation, n is the number of stages, ΔH is the total heat drop theoretically available by adiabatic expansion between the initial and terminal pressures, and E is the probable thermal efficiency of the turbine. The value of E may be obtained from the equation

$$E = \frac{2545}{S \times \Delta H}$$

In this equation, S is the probable steam consumption per horse-power per hour of the turbine.

7 In the turbine which we are designing, we have assumed the value of this efficiency to be 0.60, the heat drop to be 322.2 B.t.u. and the number of stages to be two.

8 Substituting these values we find

$$K = 0.00056 \left(\frac{2-1}{2}\right) 322.2 \, (1 - 0.60) = 0.036$$

The probable heat drop per stage will therefore be

$$\frac{322.2}{2}(1 + 0.036) = 167.0 \text{ B.t.u.}$$

Allowing this drop in the first stage, we will have for the heat content after the first isentropic expansion, 1193.3 − 167.0 = 1026.3 B.t.u. From the entropy table, the pressure of the steam entering the second stage will be 16.86 lb., since this is the pressure corresponding to the entropy 1.56 and the heat-content 1026.3. Since the efficiency of the turbine is 60 per cent, the heat transformed into work is 60 per cent of the theoretical heat drop and the heat transformed into work per stage is

$$\frac{\Delta H}{n} \times E = \frac{322.2}{2} \times 0.60 = 96.6 \text{ B.t.u.}$$

Subtracting this quantity from the initial heat content, we have for the heat content of the steam entering the second stage of the turbine 1193.3 − 96.6 = 1096.7 B.t.u. We therefore have for the entropy of the steam entering the second stage the value 1.663. Assuming the same heat drop in the second stage to be the same as that in the first stage, we will have for the heat content of the steam leaving the second set of nozzles, 1096.7 − 167.0 = 929.7 B.t.u. The pressure of steam having 929.7 B.t.u. heat content at the entropy 1.663 is found to be 1.005 lb. which gives a complete check on our work and shows the calculations to be correct.

9 If it is desired to find only the pressure of the steam as it enters each stage of the turbine, it will be found that we may proceed in the following manner: From the initial heat content of the steam, which we will designate by the symbol H_1, we will subtract the quantity

$$\frac{\Delta H}{n}\left[1 + 0.00056\left(\frac{n-1}{n}\right)\Delta H\,(1-E)\right] = h_1 \ldots\ldots [1]$$

and write

$$H_1 - h_1 = H_2 \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

From the temperature entropy table determine the pressure of steam of the initial entropy, having for its heat content H_2. This will be the pressure of the steam entering the second stage. Let us now subtract from H_2 the quantity

$$\frac{\Delta H}{n}\left[1 + 0.00056\left(\frac{n-3}{n}\right)\Delta H\,(1-E)\right] = h_2 \ldots\ldots [3]$$

and obtain

$$H_2 - h_2 = H_3 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [4]$$

The heat content, H_3, together with the initial entropy of the steam, will determine the pressure of the steam entering the third stage. The pressure of the steam entering the fourth, fifth, etc., stages is obtained in a similar manner except that we must substitute the quantities $\frac{n-5}{n}$, $\frac{n-7}{n}$, etc., for the quantity $\frac{n-1}{n}$ in [1] to obtain the quantities h_3, h_4, etc. When performing this operation in the case of any particular turbine, it will be found that the value

TABLE 1 METHOD OF DETERMINING PRESSURES

$\Delta H = 322.2 \frac{\Delta H}{n} = 53.7$ $\qquad$ $0.00056\ \Delta H\ (1-E) = 0.072$

$\frac{n-k}{n}$	$0.072 \frac{n-k}{n}$	h	H	Pressure
5/6	0.060	56.7	1193.3	164.8
3/6	0.036	55.4	1136.6	81.1
1/6	0.012	54.5	1081.2	37.9
−1/6	−0.012	52.8	1026.7	16.9
−3/6	−0.036	51.6	973.9	7.03
−5/6	−0.060	50.3	922.3	2.81
			872.0	1.00

of h is greater than $\frac{\Delta H}{n}$ for the $\frac{n}{2}$ high-pressure stages, and less than $\frac{\Delta H}{n}$ for the $\frac{n}{2}$ low-pressure stages. So far as the writer is aware, there is no particular reason why this method should give correct results, but experience shows that it does do so.

10 In order to illustrate the method of determinating pressures, we will determine the pressure in each of the stages of a six-stage turbine working between the same limits of pressure as the turbine whose design we have already considered. As before, the initial heat content is 1193.3 and the entropy 1.56. In order to make the application of the method more clear we will arrange the results in tabular form. We will assume that the efficiency, as before, is 0.60. The adiabatic drop will, of course, be 322.2 B.t.u. The work of computation is shown in Table 1.

USE OF MOLLIER'S DIAGRAM

11 The same method may be applied in graphical form to the solution of this class of problems when Mollier's total heat-entropy diagram is used. The writer believes that designers will find the following the best method to use for the graphical solution of this

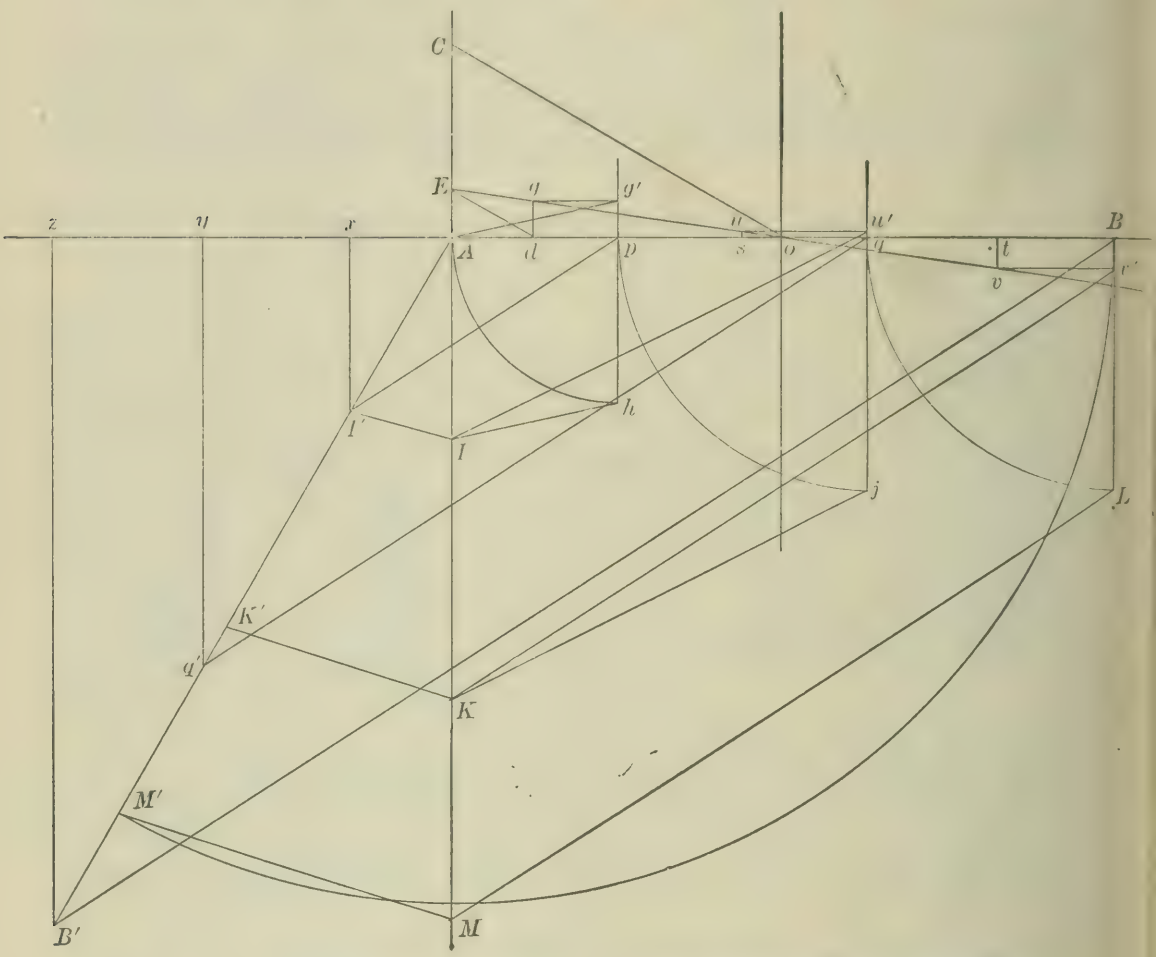


FIG. 1 GRAPHICAL CONSTRUCTION BASED ON MOLLIER'S DIAGRAM
ANGLE COA IS MUCH LARGER THAN IT SHOULD BE IN ORDER TO MAKE THE CONSTRUCTION CLEAR

problem, when the amounts of power developed by the different stages are unequal, as is sometimes the case. It is assumed that the diagram used is that accompanying the Marks and Davis steam tables. The diagram, to have real value in the designing office, should be pasted on a well-seasoned drawing board and covered with a sheet of very thin celluloid. Knowing the initial pressure and quality, or superheat, of the steam, we find on the diagram the point (*A* in Fig. 2)

representing this state, and through this point on the celluloid covering, draw a constant-heat (horizontal) and a constant-entropy (vertical) line. We will hereafter designate these two lines by the symbols H_1 and N_1 respectively. Find the state point of steam after adiabatic expansion to the exhaust pressure (H_4 in Fig. 2) by finding the intersection of N_1 with the exhaust-pressure line. The distance between the initial and final state points, of course, represents the quantity ΔH on the scale of the heat diagram. On a sheet of good detail paper draw the horizontal line AB as shown in Fig. 1, having a length equal

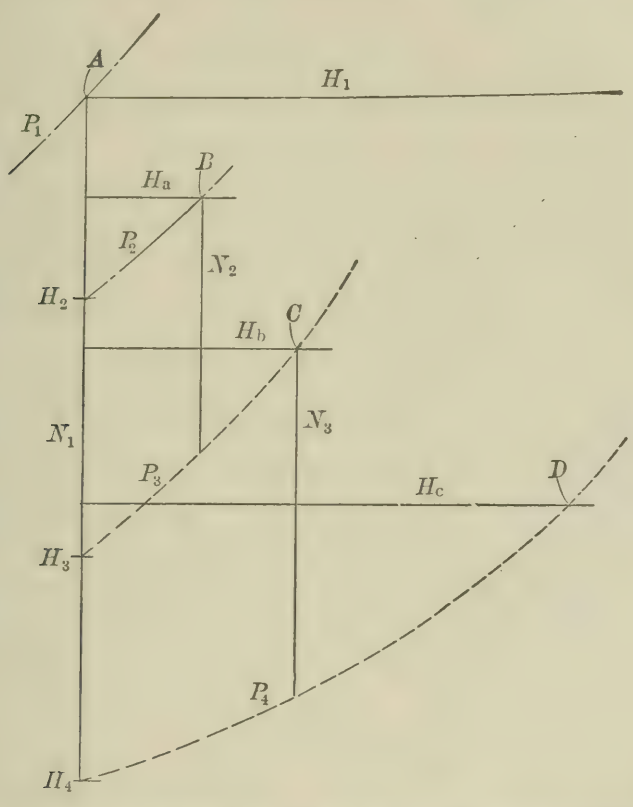


FIG. 2 SHOWING USE OF MOLLIER'S DIAGRAM

to this distance, and bisect it at O. At A, erect a perpendicular, making the length Ac equal to $0.00056\ (\Delta H^2)\ (1-E)$. This may be easily done by setting a protractor at an angle whose tangent is $0.00112\ \Delta H\ (1-E)$ and drawing line Oc through O to intersect Ac at c. Divide AB into the same number of segments as the turbine is to have stages, making the length of each segment proportional to the amount of power which is to be developed in the corresponding stage.

12 Points p and q in Fig. 1 represent the division points between the segments. Bisect each segment to locate the mid-points d, s and

t. Draw de parallel to Oc. Draw ef through O. Erect short perpendiculars at d, s and t, to intersect ef, at g, u and v. Erect perpendiculars through A, p, q and B. Draw horizontals through g, u and v to intersect the perpendiculars through p, q and B, at g', u' and v'. With p as a center, strike arc Ah. In like manner, with q and B as centers, strike arcs pj and ql. Draw Ag', and then hI parallel to it, intersecting the perpendicular through A at I.

13 In the same manner, draw jk and lm parallel respectively to $u'I$ and $v'k$. If the segments into which AB have been divided are all equal, we are now prepared to determine from Mollier's heat diagram the pressures in the steam chests of the several stages in the following manner: Through lines N_1 on the celluloid covering of the heat diagram, draw short lines, H_2, H_3, and H_4 as shown in Fig. 2, making the distances of these several lines from line H_1 equal to AI, AK, and AM respectively. The intersections of these lines with line N_1 will determine on the heat diagram, the pressure of the steam in the steam chests of the several stages, or in the exhaust. If the diagram is correctly drawn, AM will be equal in length to AB, and the exhaust pressure found will be that originally fixed upon.

14 If, however, the segments are of unequal length for any reason, as would be the case if the diameters of the several rotating members are unequal, we will find that AM would not be equal to AB, and the exhaust pressure will be incorrect. In order to determine the pressure of the steam entering each stage under these conditions, we must alter our diagram in the following manner: About A as a center, strike an arc in the manner indicated, having the length AM' equal to AB. Through point A, with a 60-deg. triangle, draw AM' intersecting the arc at M'. Draw MM', and then draw KK' and II' parallel to it. By following the same procedure as was outlined in the preceding paragraph, but using distances AI', AK' and AM' to locate H_2, H_3 and H_4, we may readily determine the correct value for the pressure of the steam entering any stage of the turbine.

15 If it is desired to determine from the diagram all of the properties of the steam as it enters each stage of the turbine, and to check the work, we may proceed as follows: Note that AI' is the actual heat-drop in the first stage, and that the actual heat-drop in each succeeding stage bears the same proportion to its segment as AI' does to the segment Ap. We may accordingly determine the actual heat drop in each stage by drawing pI', and then drawing qq' and BB' parallel to it, intersecting AM' at q' and B'. We will now have AI' equal to the actual heat-drop in the first stage, $I'q'$ equal to the actual heat drop

in the second stage and $q'M'$ equal to the actual heat-drop in the third stage. We may complete the design in the following manner: Lay off to the left a distance Az equal to $E \times AB$, where E is the efficiency of the turbine expressed as a fraction. Draw $B'Z$, and parallel to it $q'y$ and $i'x$. On the celluloid covering of the diagram draw line H_a parallel to H_1 and at the distance Ax below it. Through the point B, determined by the intersection of H_a and the pressure line P_2 determined by H_2, draw line N_2 parallel to N_1. The point B is the state point of the steam entering the second stage. From this point measure downward the distance $I'q'$ on the line N_2. The point so found should fall or the pressure line P_3, determined by H_3. Draw H_b parallel to H_1 and distant Ay from it, and find its intersection at C with the pressure line P_3, determined by H_3. This will be the state point of the steam entering the third stage. Through it draw N_3 parallel to N_1, and from it measure downward on N_3 the distance $q'B'$. The point so obtained should fall on the exhaust-pressure line. Draw H_c distant Ax from H_1 and find its intersection at D with the exhaust-pressure line. This intersection is the state point of the exhaust steam.

16 It is believed that after the designer has once familiarized himself with the method of constructing this diagram, the time required to perform carefully and accurately the directions given and to make all the determinations necessary will be comparatively short, certainly not more than an hour in the case of a four-stage turbine, and that the method will eliminate much of the tedious labor which is necessary when the method of successive approximation is used in performing this work.

DISCUSSION

W. H. Herschel (written). Referring to Fig. 1 the total available heat is not $AB = \Delta H$, but AB', and it is the main object of the paper to determine the value of AB'. The method is based upon that of Professor Peabody, and differs from it in leaving out all consideration of temperature so as to permit the use of the Marks and Davis entropy diagram. We are therefore concerned only in proving whether the accuracy of the method is as great as that attainable in the customary use of the diagram.

The proposed graphical method is open to the objection that the long line BB' must be drawn parallel to the much shorter line pI'. This cause of inaccuracy may be removed by computing and laying off the length AB', and then drawing qq' parallel to BB'. To serve as a check qq' should also be parallel to pI'.

From the graphical construction $AB' = \frac{AB^2.AI}{Ap.AM'}$, where $AM = AB + (us + dg - tv)$. The quantity in parenthesis is the sum of the intercepts between the lines EF and AB ordinates above AB being plus, and those below AB minus, all these ordinates being proportional to their distances from O. To make the equation

$$od + os + (-ot) = \frac{nAB}{2} - (n-\tfrac{1}{2})\,Ap - (n-1\tfrac{1}{2})pq - (n-2\tfrac{1}{2})\,Bq$$

more general, we may designate the segments of the line AB by the symbols S_1, S_2, etc., and then

$$od + os + (-ot) = \frac{n\,\Delta H}{n} - (n-\tfrac{1}{2})\,S_1 - (n-1\tfrac{1}{2})\,S_2 \ldots\ldots [n-(n-\tfrac{1}{2})]\,S_n$$

$$us + dg - tv = \frac{[od + os - ot]\ \mathrm{AE}}{\frac{\Delta H}{2}}$$

$$AM = \frac{od + os - ot}{\frac{\Delta H^2}{2\,(AC)\,S_1}} + AB$$

and finally

$$AB' = \frac{\Delta H^2 \left[S_1 + S_1 \frac{AC}{(\Delta H)^2}(\Delta H - S_1)\right]}{S_1 \left\{\Delta H + \frac{\frac{n\,\Delta H}{2} - (n-\tfrac{1}{2})S_1 - (n-1\tfrac{1}{2})S_2 \ldots\ldots [n-(n-\tfrac{1}{2})]S_n}{\frac{(\Delta H)^2}{2(AC)\,S_1}}\right\}} \quad \ldots[1]$$

Equation [1] looks somewhat formidable, but is in reality simple to use, since it contains only n, the number of parts into which ΔH is divided, the lengths of each of these parts, and a constant $AC = 0.00056\ (\Delta H^2)\ (1-E)$, where E is what Professor Peabody calls the overall internal heat factor. If AB is divided into equal parts, $AB' = nAI = nh_1$.

According to the proposed modification of Professor Peabody's method

$$h_1 = \frac{\Delta H}{n}\left[1 + 0.00056\ (1 - E)\ \Delta H \left(\frac{n-1}{n}\right)\right]$$

and

$$h_1 - h_2 = \frac{2(\Delta H)^2}{n^2}\,0.00056\ (1 - E)$$

Referring to Fig. 2, we will call B_2 the point where the perpendicular dropped from the point B, intersects the line of constant pressure P_3. If the power to be developed in each stage is the same, $h_1 = AH_2 = BB_2$. Because it is fairly obvious that the method under consideration does not apply to the superheated steam part of the diagram, and because it is customary to design turbines for dry steam, I shall confine my attention to the part of the diagram below the saturation curve, where the pressure lines are substantially straight. Let Θ_1 = the angle between the line BH_2 and the vertical AH_2. Then in order that AH_2 shall equal BB_2, $\frac{BB_2 - H_2H_3}{\tan \Theta_1 h_1 (1-E)}$ = rate of divergence of lines P_2 and P_3 in the horizontal distance between the verticals N_1 and N_2

$$= \frac{\frac{2(\Delta H)^2}{n^2} 0.00056\ (1 - E)}{\tan \Theta_1 \left\{ \frac{\Delta H}{n} \left[1 + 0.00056\ (1 - E)\ \Delta H \left(\frac{n-1}{n} \right) \right] \right\} (1 - E)}$$

$$= \frac{2 \Delta H\ 0.00056 \cot \Theta_1}{n \left[1 + \left(\frac{n-1}{n} \right) 0.00056\ \Delta H\ (1-E) \right]}$$

It should be noted that the horizontal distance between N_1 and N_2 must not be measured in the scale of entropy, but in the scale of H, 1 in. = 50 B.t.u. But if Θ_2 is the angle between the lines N_1 and P_3, then the rate of divergence of lines P_2 and P_3 is equal to $\cot \Theta_1 - \cot \Theta_2$ and we have

$$\cot \Theta_1 - \cot \Theta_2 = \frac{2\ \Delta H\ 0.00056 \cot \Theta_1}{n \left[1 + \left(\frac{n-1}{n} \right) 0.00056\ \Delta H\ (1 - E) \right]} \quad \ldots [2]$$

The rate of divergence is thus seen to depend on the quantities ΔH, E, n, and $\tan \Theta_1$, and under ordinary conditions their variation will be about as follows:

ΔH from 350 to 140 B.t.u.
E from 0.50 to 0.70
n from 2 to 25 to include the Rateau type, but not the single-stage De Laval
Θ_1 from 55 deg. at 200 lb. to 65 deg. at 1 lb.
cot Θ_1 from 0.700 to 0.466

Equation [2] will give a maximum value when $\cot \Theta_1 = 0.700$; $n = 2$; $\Delta H = 350$ and $E = 0.70$; and a minimum value when $\cot \Theta_1 = 0.466$; $n = 25$; $\Delta H = 140$ and $E = 0.50$. This gives for the maximum divergence in 8 in. horizontal length, 53.4 B.t.u. and for the minimum divergence for the same difference of entropy, 1.13 B.t.u.

If we restrict the formula to land Curtis turbines, n would not have a maximum value above 6 (the highest number of stages mentioned by the author), and the minimum divergence would become 4.72. If we leave out low-pressure and non-condensing machines, ΔH would not be less than 250 and the minimum divergence would be 8.56. I have thus far treated ΔH, E, n and $\cot \Theta_1$ as independent variables, following the example of the author, who assumes the same efficiency for a two-stage as for a six-stage turbine with the same value of ΔH. However as E depends upon bucket speed as well as on $\frac{\Delta H}{n}$ and as the bucket speeds are not given, we may assume that with suitable speeds and types of machines, the two turbines might have the same efficiency.

Professor Peabody applies his method to a somewhat wider range of conditions than does the author, but in order that the latter may not make the objection that I attempt to apply it to conditions for which it was not intended, I shall confine myself to the two cases which he has himself given, where $\Delta H = 322.2$ and $E = 0.60$ in both cases and $n = 2$ and 6.

Where $n = 6$, Θ_1 varies from 58 deg. to 64 deg. and we may take the average value of $\cot \Theta_2$ to be 0.546 where $n = 2$, $\Theta_1 = 62$ deg. and $\cot \Theta_1 = 0.532$. Therefore, we have from [2]

$$\left.\begin{array}{l}\text{Divergence in 8 in.} \\ \text{if } n = 2\end{array}\right\} = \frac{2 \times 322.2 \times 0.00056 \times 400 \times 0.532}{2\,[1 + \frac{1}{2}\,(0.00056)\,322.2\,(1 - 0.60)]} = 37.1 \ldots [3]$$

$$\left.\begin{array}{l}\text{Divergence in 8 in.} \\ \text{if } n = 6\end{array}\right\} = \frac{2 \times 322.2 \times 0.00056 \times 400 \times 0.546}{6\,[1 + \frac{5}{6}(0.00056)322.2\,(1-0.60)]} = 12.4 \ldots [4]$$

Now it is an easy matter to find the actual divergence on the entropy diagram in 8 in. horizontal distance and for a drop of say 2 in. = 100 B.t.u. If we measure the distance apart of two pressure lines at entropy 1.30 and again at 1.46, the difference, to the scale of 1 in. = 50 B.t.u., will give us the divergence in B.t.u. for 4 in. horizontal distance, and double this would be the rate of divergence of those particular pressure lines in 8 in. In the case of the lower pressures the full 8 in. may be measured directly, as from entropy 1.46 to 1.78,

the horizontal entropy scale being 1 in. = 0.04 units. In this way I find the actual divergence in 8 in. horizontal and 2 in. vertical is about 25 to 35 B.t.u. in the most used part of the diagram and has about double those values for very high pressures.

In the author's problem of the two-stage turbine, the entropy at admission is 1.560 and for the second stage 1.663. The horizontal range is thus $\frac{1.663 - 1.560}{0.04} = 2.57$ in. From [3] we might therefore expect a divergence of $\frac{2.57 \times 37.1}{8} = 11.9$ B.t.u., while the actual divergence as found by the author is 167.0 − (1026.3 − 871.1) = 11.8. A divergence of 11.8 for a horizontal distance of 2.57 in. and a vertical drop of $\frac{1026.3 - 871.1}{50} = 3.10$ in., corresponds to a divergence of 24 B.t.u. for our standard conditions, instead of 25 to 35 B.t.u. which I found by scaling from the diagram. Another estimator might find a different value by scaling, but even if we suppose 30 to be correct, this would not change very greatly the available heat per stage. The divergence for our two-stage turbine would become 14.8 and the drop would be 168.5 instead of 167.0. This would be equivalent to increasing the factor 0.00056 to 0.00071. As the steam velocity $= C\sqrt{h}$, an increase of h from 167.0 to 168.5 would increase the steam velocity only about half of one per cent. Since the proper steam velocity for a given bucket speed is not known within half of one per cent, we may conclude that the proposed method is accurate enough for a two-stage turbine working under moderate pressure.

Coming now to the six-stage turbine, the entropy at admission is 1.560 as before. The entropy of the last drop is not stated but it may be found as follows: The drop in each stage is $\frac{\Delta H}{n}(1+K) = 56.9$, so that the drop from A to D (Fig. 2) would be 0.60 (6 × 56.9) = 204.8 and the drop from A to P_{n+1} (P_4 on Fig. 2, or P_7 for the six-stage turbine) would be 204.8 + (56.9 × 0.40) = 227.6. The end of the last drop would be at H = 965.7 and pressure = 1.00, and the entropy of this point would be 1.729. The horizontal range would be $\frac{1.729 - 1.560}{0.04} = 4.22$ in. and so we should expect from [4] a divergence of $\frac{4.22 \times 12.4}{8} = 6.5$ or 1.3 B.t.u. for each of the five drops. This should be equal to $h_1 - h_2 = 56.7 - 55.4$ in Table 1. The

author takes $\frac{322.2}{6} = 53.5$ instead of 53.7, but allowing for this inaccuracy, $h_1 - h_2 = 56.9 - 55.6 = 1.3$ as before.

The author says, Par. 9: "So far as the writer is aware there is no particular reason why this method should give correct results, but experience shows that it does do so." In Par. 6 he uses the word empirical which is the only indication he furnishes that his method is not applicable with perfect accuracy to all parts of the entropy diagram. As I have already pointed out, the method does not apply to superheated steam or to wet steam of very high pressure, and in regard to the part of the diagram in most frequent use, it would be more accurate to say the error is so small as not to be readily detectable, than to leave it to be inferred that there is no error.

The author has carefully omitted all references made by Professor Peabody to errors in the method. For example, Professor Peabody's Table 6* (corresponding to the author's Table 1) contains the following data:

Pressure	Entropy	Heat Drop	
		1193.3	
		1136.7	
164.8	1.560		56.6
		1158.4	
		1101.5	
80.1	1.588		56.9
		1123.5	
		1066.4	
38.1	1.618		57.1
		1088.6	
		1031.9	
16.9	1.651		56.7
		1053.7	
		996.7	
7.09	1.685		57.0
		1018.8	
		962.1	
2.76	1.722		56.7
1.00	1.761	Sum	...341.0

in reference to this tabulation Professor Peabody remarks that the

* A Method of Determining Pressures for Steam Turbines, Cecil H. Peabody, Trans. Soc. Naval Archs. and Marine Engrs., 1909, p. 32.

sum of the several amounts of heat assigned to the six stages for adiabatic action should be equal to $322.2 \times 1.054 = 339.6$, instead of 341.0, the discrepancy being due to lack of precision in the calculation.

In the case of the two-stage turbine Professor Peabody assigned a drop of 166.4 B.t.u. to the first stage and obtained 166.1 for the second. In short, even with the most careful interpolation in his entropy table Professor Peabody found that his method was subject to error, though to be sure the error in the cases cited is negligible for all practical purposes.

Returning to the author's statement in Par. 9, we are left in doubt as to the extent of the "experience" which shows that the method gives "correct results." There is nothing to show that the experience covers more than the examples given in Professor Peabody's and the author's papers. However, if it is desired to apply the method to any part of the diagram, an idea of the error involved may be obtained by the help of [2], either by scaling the divergence of the lines P_2 and P_3 or by measuring with a protractor the angles Θ_1 and Θ_3. For this purpose the location of the lines P_2 and P_3 may be obtained with sufficient accuracy if AH_3 is taken equal to $H_2H_3 = \dfrac{\Delta H}{n}$.

But the author's method, in addition to assuming a constant rate of divergence for a certain drop and a certain difference of entropy, is based on other assumptions which affect its usefulness. In Fig. 1 it will be noted that $B'z$, $q'y$ and $I'x$ are parallel, which signifies that the efficiency of each stage is the same, and Professor Peabody says: "The method assumes that the efficiencies of the several stages are the same or nearly the same, and most concordant results will be obtained when the same amount of heat is assigned to each stage. But the method is insensitive to reasonable changes in the heat factor and will give good comparison with practice with any type of turbine."[1]

That the method is insensitive to changes in E is shown by Professor Peabody's calculations, and it also follows from [2]. When Professor Peabody says, "any type of turbine," it is to be supposed he means any multicellular turbine, since the method does not apply to a single-stage De Laval machine, and its application to the Parsons type has not been demonstrated. As for the comparison with "prac-

[1] A Method of Determining Pressures for Steam Turbines, Cecil H. Peabody, Trans. Soc. Naval Archs. and Marine Engrs., 1909, p. 30.

tice," it depends upon what is meant. Writers show a "strange reticence" in telling all they know and it is a question whether practice is to be considered as represented by what is published, or by the methods actually used by designers.

I question whether it is good practice to assume a constant-stage efficiency as it is obviously contrary to fact. As the efficiency can only be constant if the losses are constant, we must consider the losses in each stage. Leaving out radiation, leakage between stages and the loss due to residual velocity, $\frac{w_2^2}{2g}$, the chief losses are due to the steam passing through the nozzles and blades, and to the disks and blades rotating in steam. I shall call these two frictional losses, blade losses and windage.

If c = the theoretical velocity due the heat drop assigned to a stage, $c_1 = \varphi c$ = velocity of steam leaving the nozzle, and $w_2 = \chi w_1$ where w_1 is the resultant of c_1 and the peripheral velocity. Then the blade loss will equal

$$\frac{1}{2g}[(c^2 - c_1^2) + (w_1^2 - w_2^2)] = \frac{1}{2g}[c^2(1 - \varphi^2) + w_1^2(1 - \chi^2)]$$

As φ and χ may be considered constant, it follows that the blade loss will be constant only when c and w_1 are the same for all stages. If the heat assigned to a stage is constant, c is constant, but w_1 will vary with the blade angle. It is well known that at the low-pressure end of marine turbines, semi-wing, wing and double-wing blades are employed, and this same flattening of the blade takes place in land turbines, though perhaps not so often referred to. Consequently the blade losses vary with changes in the value of w_1 from stage to stage.

The variation in the windage losses is much greater. While the absolute numerical value of the windage losses may be difficult to discover, on account of disagreement between authorities whose tests were made with different types of running wheels, the following equation from Stodola[1] may be taken to illustrate our point:

$$N = \beta 10^{-9} D n^3 L \gamma$$

where

N = windage in kw.
β = a factor depending upon the number of rows of blades
D = diameter to middle of blade in m.
L = mean length of blade in m.
γ = specific weight of steam

[1] The Steam Turbine, Aurel Stodola, ed. 4, p. 129.

D may be constant in all stages, and in some cases β is also a constant. L may vary from a fraction of 1 in. to over a foot, and the variation in the density of steam is enormous. Consequently the windage will be greatest in the first stage, where γ is a maximum, in spite of the fact that L is there a minimum, and it will gradually decrease to a comparatively insignificant quantity in the last stage. Of course, if there are very few stages, or the turbine is running non-condensing, the windage in the last stage may be considerable. The assumption that the efficiency is the same for each stage is thus far from the truth, and can be justified only on the ground that it is one commonly made in books.

In Par. 6, the author defines E as the probable thermal efficiency of the turbine and gives the formula $E = \frac{2545}{S \times \Delta H}$ where S is the probable steam consumption per horsepower per hour. In using this formula, how he accounts for the fact that there are losses which do not increase the entropy of the steam, does not seem clear. Stodola[1] explains the matter more fully somewhat as follows: Add to N_e the effective horsepower, the losses outside the turbine casing, such as bearing friction and the power used to drive auxiliaries, and the sum is called N_i. If G_e is the steam consumption per effective horsepower per hour, then $G_i = G_e \frac{N_e}{N_i}$, the difference in total heat between initial and final conditions will be $\frac{2545}{G_i}$ (neglecting losses due radiation and the increase in steam velocity between the steam chest and the exhaust), and finally the efficiency will be equal to $\frac{2545}{G_i \Delta H}$. Professor Peabody uses practically this same method, except that he takes $G_i = 0.9\ G_e$. He then goes on to say: "There may be a reasonable dislike to using a factor based on the gross horsepower of the turbine, but it will be shown that a considerable variation in that factor will have only a small effect on the computations of pressure; and after all there appears to be no way to avoid the difficulty."

As I understand Professor Peabody, the heat drop between the point A and the point D (Fig. 2) is equal to $\frac{2545}{G_i}$ and there is no objection to this method of finding the point D. But he does admit

[1] The Steam Turbine, Aurel Stodola, p. 116.

there might be a reasonable dislike to taking the overall turbine efficiency, $\frac{2545}{G_1 \Delta H'}$, as the efficiency of each stage.

It might also be objected that the steam does not actually pass through the changes represented by the broken line AH_2B, etc. This could occur only if there were no friction and consequently no increase of entropy in the nozzles and no decrease of pressure in passing through the blades. In reality there is nozzle friction and especially in Curtis turbines using intermediates or reversing blades, there is often a fall in pressure in passing the blades. On this account some writers do not classify the Curtis turbine as a pure impulse turbine. However, the error introduced by considering that the steam follows the path AH_2B is slight compared with that due to taking E constant. An inaccurate value of E has, as Professor Peabody shows, but slight influence on the calculated steam velocities, but it should not be overlooked that its influence on the calculated steam volumes is much greater.

However the total amount of heat that may be assigned to the several stages is obtained, this is but a beginning. The real task is in so arranging blade lengths, blade angles, disk diameters, heat drops, etc., that the ratio of steam speed to bucket speed shall be as desired. Professor Peabody[1] and the author (Par. 3) both assume a constant steam velocity and the same heat drop in each stage. Now the theoretical ratio between bucket and steam velocity in an impulse turbine is one-half and if there were no losses this value would be used. On account of the losses, which increase approximately as the cube of the bucket speed, there is a certain bucket speed less than half the steam speed, which gives the greatest net power. Furthermore, not only has a turbine a most efficient speed of revolution, but each separate row of blades has a most efficient linear velocity depending on the losses in the separate stages. The angular velocity of every row of blades must be the same, but the linear velocity may be regulated within certain limits, by changes in blade length and disk diameter. Thus if the author assumes that the diameters of the moving elements of each stage are the same, he should not assume a constant steam velocity in each stage.

The method presented by the author gives results that agree with those obtained by the method of assuming a constant reheating

[1] A Method of Determining Pressures for Steam Turbines, Cecil H. Peabody, Trans. Soc. Naval Archs. and Marine Engrs., 1909, p. 29.

factor, with an error so small as not to be readily detectable by scaling on the entropy diagram, so long as operations are confined to the more usual pressures, with steam initially dry. However, the reheating factor actually varies from stage to stage with changes in the internal losses, so that any method which assumes a constant stage efficiency is but a preliminary to the more difficult task of proportioning the pressure drop between the stages so as to obtain a suitable variation in the ratio of bucket speed to steam velocity, which can be accomplished only by the method of successive approximations.

C. H. PEABODY[1] (written). In his paper Profesor Cardullo shows that his empirical equation is both accurate and convenient, and the few applications of it which I have made lead to the same conclusion. But I cannot agree that in practice it will be found either more simple or exact than the method offered in my paper.[2] Though it does not much change conditions, I will say that the computations of ratios for determining temperatures and pressures have been abbreviated since my paper was presented.

I should like to explain the derivation and use of these factors. Taking 164.8 lb. and 1 lb., or 366 deg. and 102 deg., Professor Cardullo's paper shows that the middle temperature for adiabatic action is 223.5 deg. But if only 0.6 of the available heat is changed into work and the other 0.4 remains in the steam, the entropy will increase to about 1.65. Let h stand for the heat contents and find the ratio $\frac{\Delta h_{1.56}}{\Delta h_{1.65}}$ for a range of temperature of 40 deg. at entropy 1.56 and at entropy 1.65 as follows:

Temperature	Heat Contents	
	Entropy 1.56	Entropy 1.65
243	1056.0	1122.8
102	1006.7	1069.6
	49.3	53.2

Ratio: $\frac{53.2}{49.3} = 1.08$

To find intermediate temperatures for a six-stage turbine, draw a diagram like Fig. 3 in which h_1h_2 represents the adiabatically avail-

[1] Prof. Naval Arch. and Marine Engrg., Mass. Inst. of Tech., Boston, Mass.

[2] A Method of Determining Pressures for Steam Turbines, Trans. Soc. Naval Archs. and Marine Engrs., 1909, p. 32.

able heat, 322.2 B.t.u. At a convenient scale the ratio 1.080 is laid off at the top and a diagonal line is drawn through the middle point. The line h_1h_2 is divided into six equal parts. There may in any case be as many parts as there are stages which may be equal or unequal. At the middle points of the segments of h_1h_2 draw abscissae and on them read their values as indicated. Then calculate the intermediate

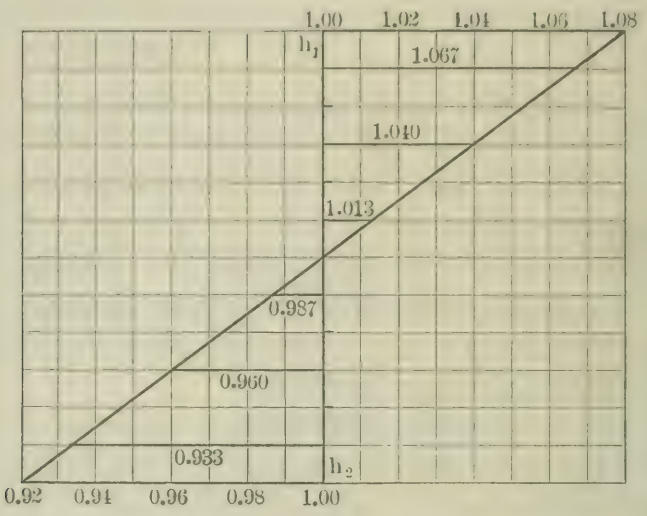


FIG. 3 DIAGRAM SHOWING INTERMEDIATE TEMPERATURES FOR A SIX-STAGE TURBINE

temperatures and pressures as follows:

$$\frac{322.2}{6} = 53.7$$

		Temperature	Pressure
$53.7 \times 1.067 = 57.3$	$1193.3 - 57.3 = 1136.0$	312.5	80.5
$53.7 \times 1.040 = 55.9$	$1136.0 - 55.9 = 1080.1$	263.5	37.6
$53.7 \times 1.013 = 54.4$	$1080.1 - 54.4 = 1025.7$	218.0	16.5
$53.7 \times 0.987 = 53.0$	$1025.7 - 53.0 = 972.7$	176.5	6.95
$53.7 \times 0.960 = 51.5$	$972.7 - 51.5 = 921.2$	138.0	2.74
$53.7 \times 0.933 = 50.1$	$920.2 - 50.1 = 871.1$	102.0	1.00
322.2			

The temperatures and pressures are looked up in the temperature-entropy table at entropy 1.56, interpolating to half degrees, but not to a greater degree of refinement. Great refinement is wasted in this matter, for the straight adiabatic method of text books gives

a maximum error in this case about 0.06 error in energy, which corresponds to 0.03 error in velocity. A variation of 10 per cent from the normal velocity of a turbine has a barely appreciable effect on the efficiency, and a variation of 0.015 would be imperceptible either in computation or experiment.

There is, however, no use in allowing a preventable error of 3 per cent to creep into one computation and derange the results, consequently some method of correction should be used. I propose the ratio in Table 2 for my method.

TABLE 2 RATIO FOR PRESSURE DISTRIBUTION

Heat Factor	150 Gage Pressure to 28 in. Vacuum	150 Gage Pressure to Atmosphere	Atmosphere to 28 in. Vacuum
0.55	1.09	1.050	1.040
0.60	1.08	1.045	1.035
0.65	1.07	1.035	1.030
0.70	1.06	1.030	1.025
0.75	1.05	1.025	1.020

Having the ratio of heat changed into work by the turbine to the total available heat for adiabatic expansion, the designer may select the nearest ratio from the table and with it draw a diagram like Fig. 3 and complete the distribution of heat as shown by Table 2. The ratio should be read to three decimal places because any error there may be in the method of distributing pressure is spread over the entire range of temperature and the intervals of temperature and pressure are obtained with a desirable degree of regularity.

To show the insensitiveness of the method we may note that variation in the ratio of heat changed into work to the total available heat is as follows:

Heat factor	0.55	0.65	0.75
Intermediate temperature	218	219	220
Intermediate pressure	16.5	16.9	17.2

And further to show that a considerable variation in pressure has but a slight effect we may compare results with a ratio selected from Table 2 and with the proper ratio for the range of pressures:

Heat factor 0.65	Tabular Value	Special Calculation
Initial temperature	366	400
gage pressure	150	232
Final temperature	102	102
vacuum, in.	28	28
Ratio	1.07	1.075
Intermediate temperature	231.8	232.1
pressure absolute	21.49	21.61

While I have no question that Professor Cardullo can use his graphical method effectively, I remain of the opinion that a numerical computation with the aid of the temperature-entropy table will be found more accurate and rapid.

W. D. Ennis (written). The simplicity and exactness of Professor Cardullo's method lie in the use of the corrective factor K which he

TABLE 3 HEAT DROP IN SIX-STAGE TURBINE

Absolute Pressure, Lb. per Sq. In.	Entropy	Heat Contents, B.t.u.
160	1.629	1252 0
81	1.661	1216.8
39	1.6935	1181.6
17.3	1.731	1146.4
7.3	1.772	1111.2
2.9	1.814	1076.0
1.0	1.863	1040.8

has discovered. In designing a multi-stage turbine, either some inequality in the heat drops for the different stages or a rather prolonged adjustment must be contemplated. There seems to be no clear reason for the form of this new corrective factor and one is inclined to doubt the generality of application of such expressions. I have, however, found that it fits a set of conditions other than those given in the paper and am prepared at least to try it in additional cases.

Consider a turbine having an initial pressure of 160 lb. abs., with 100 deg. of superheat, expanding down to 1 lb. abs., six stages, steam consumption 12 lb. per h.p-hr.: The initial heat content is 1252, the final (after adiabatic drop) 911; making the heat drop = 341 B.t.u., the efficiency $E = 0.62$, and $K = 0.0608$. The ideal heat drop per stage, corrected by the K factor, is then 60.2 B.t.u.; the actual drop is $\frac{341}{6} \times 0.62 = 35.2$ B.t.u. This leads to Table 3.

The whole heat drop is very nearly $1252 - 1040.8 = 211.2 = 341 \times 0.62$. It would be interesting to know how the expression for K was obtained and whether it has further justification than the one that it "works," which is of course excellent.

THE AUTHOR. The objections raised by Mr. Herschel may be classified as follows:

a That the method proposed will not give satisfactory results when wet steam is used in the turbine.

b That it will not give satisfactory results when superheated steam is employed.

c That it assumes the losses to be equal in each of the stages.

d That it assumes all of the losses to be of such a character as to increase the entropy of the steam.

The answer to the first objection, namely that the method is not satisfactory when applied to wet steam, may be seen in Table 4, in

TABLE 4 FOUR-STAGE TURBINE

INITIAL STEAM PRESSURE 14.70 LB. ABS., INITIAL QUALITY 85 PER CENT, TERMINAL PRESSURE 0.594 LB. ABS., ASSUMED EFFICIENCY 70 PER CENT.

$$\triangle H = 168.8 \frac{\triangle H}{n} = 42.2$$

$$0.00056 \triangle H (1 - E) = 0.028$$

$\frac{n-k}{n}$	$0.028 \frac{n-k}{n}$	h	H	Pressure	Actual		
					Entropy	Heat Drop	Heat Content
0.75	0.021	43.1	1004.6	14.70	1.54		1004.6
			961.5	7.19	1.54	43.1	961.5
0.25	0.007	42.5			1.56		974.45
			919.0	3.278	1.56	43.1	931.35
0.25	0.007	41.9			1.582		944.2
			877.1	1.426	1.582	43 1	901.1
0.75	0.021	41.3			1.604		913.05
			835.8	0.594	1.604	43.1	869.95

which will be found the calculations for a turbine taking steam of 85 per cent quality and expanding it from a temperature of 212 deg. to a temperature of 85 deg. This is an extreme case and there is no reason why a turbine should be supplied with steam of this quality even though the steam rejected by the engine unit is of 85 per cent quality. The use of a separating receiver between the engine and the turbine would eliminate the water without appreciably decreasing the admission pressure to the turbine. It will be seen from Table 4 that the calculation by this method of the pressure and energy drops in the different stages give results which check absolutely.

Mr. Herschel's objection that the method does not apply to superheated steam is valid, although a much closer approximation to the proper pressure and energy drops per stage may be obtained by employing this method than by the assumption of adiabatic expansion. In Table 5 will be found the calculations for a very extreme case. The turbine is assumed to take steam at 215 lb. pressure and 226 deg. superheat and to expand it to atmospheric pressure. It will be seen there that the calculated energy drop per stage is 59.3 B.t.u., while the actual energy drop in the first stage is 59.3 B.t.u., in the second, 60.2, in the third, 62.8, and in the fourth 59.3 B.t.u. By employing the method, the energy drop per stage is much more nearly equalized than it would be otherwise.

TABLE 5 FOUR-STAGE TURBINE

INITIAL STEAM PRESSURE 215 LB. ABS., INITIAL SUPERHEAT 225 PER CENT, TERMINAL PRESSURE 14.70 LB. ABS., ASSUMED EFFICIENCY 70 PER CENT.

$$\triangle H = 230.6 \quad \frac{\triangle H}{n} = 57.65$$

$$0.00056 \triangle H \ (1 - E) = 0.039$$

$\frac{n-k}{n}$	$0.039 \frac{n-k}{n}$	h	H	Pressure	Actual		
					Entropy	Heat Content	Heat Drop
			1322.5	215	1.67		
0.75	0.029	59.32	1263.2	122.7	1.67	1263.2	59.3
					1 6885	1281.0	
0.25	0.010	58.22	1205.0	66.0	1.6885	1220.8	60.2
					1.71	1238.9	
0.25	0.010	57.08	1147.9	31.97	1.71	1176.1	62.8
					1.7345	1195.0	
0.75	0.029	55.98	1091.9	14.70	1.7345	1135.7	59.3

It must be remembered that this method is in the nature of a rule of thumb for determining as closely as possible the pressure of the steam as it enters the several stages of a turbine, on the assumption that the energy drop is the same in every stage. It is intended as a preliminary to the actual design of the steam passages and the vanes of the several stages of the turbine. In order to obtain a workable rule the difference in the efficiencies of the several stages was disregarded. Had this difference been taken into account the method would become tremendously complicated and the final results practically identical with those obtained by employing the method in its simplified form. While the method is theoretically incorrect and inexact, in practice it is much more accurate than is our knowledge

of the properties of steam or the efficiencies of turbines, and therefore as accurate as is necessary or desirable.

The only losses in a steam turbine which do not increase the entropy are those due to bearing friction and radiation. In any case these are an exceedingly small fraction of the total heat supplied to the turbine and are therefore negligible in comparison with the losses which increase the entropy of the steam. To separate these losses is impossible and unnecessary and, as pointed out by Professor Peabody, slight changes in the assumed efficiency of the turbine have almost no effect in changing the value of the factor K in the equation.

I agree with Professor Peabody that a numerical computation by the aid of the temperature entropy table is more accurate than the employment of the Mollier diagram and that, in general, analytical methods are to be preferred to graphical ones for work of this kind.

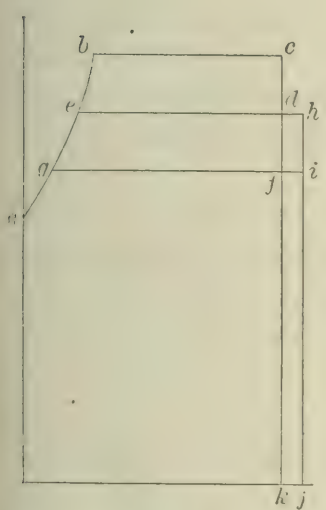


FIG. 4

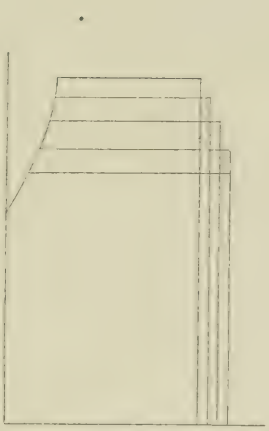


FIG. 5

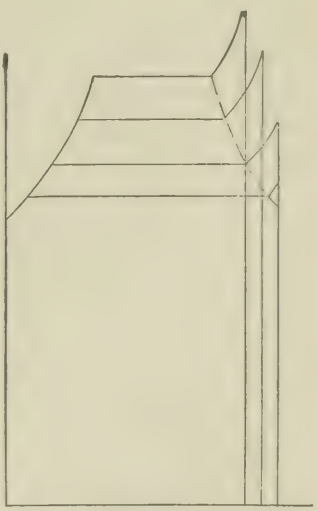


FIG. 6

DIAGRAMS ILLUSTRATING EFFECT WITH WET STEAM AND SUPERHEATED STEAM

Professor Ennis shows that when the amount of superheat is not extraordinarily high, the method gives satisfactory results. On the other hand, for extreme conditions of superheating, it is not so exact.

Why the method works is shown in Fig. 4 which represents the temperature-entropy diagram of a turbine employing wet steam as a working fluid. The line *ed* is drawn so that the area *gbcf* is divided into two equal parts, in order that there shall be an equal distribution of

energy in the two stages. Then the heat transformed into work in the nozzles of the first stage is *ebcd*. Of this a certain fraction is retransformed into heat and passed on to the second stage, this heat being represented by the area *dhjk*, of which *dhif* is retransformed into work in the second stage, and the total quantity of heat transformed into work in both stages is *gbcdhi*. It will be seen that as the total heat drop increases, the proportion of the heat supplied which is transformed into work also increases, so that the quantity *dhif* becomes a larger and larger proportion of *dhjk*, as the total heat drop increases. The extra amount of heat added per stage is therefore proportional to the total heat drop. By increasing the number of stages to four, we will get the condition of affairs shown in Fig. 5, in which the shaded areas are added as well as the areas already added in Fig. 4. Hence we must introduce the factor $\frac{N-1}{N}$ in obtaining the value of K. The amount of heat added to the steam entering the second stage by the inefficiency of the first stage is of course proportional to one minus the efficiency of that stage; hence we have the factor $1-E$. Finally the factor 0.00056 is obtained by trial and adjustment.

Fig. 6 shows why the method is not exact when applied to superheated steam. It might be made exact by introducing a factor whose value would depend upon the degree of superheat of the steam, but this introduces a complication, which destroys the simplicity of the method, and which is therefore objectionable. Rather than complicate the method, we may calculate the heat drops in the several stages as in Table 5, take their average, and employ that as the heat drop per stage. The average of the actual heat drops is 60.4. The computation of the proper pressures for each stage may be made accordingly.

No. 1318

THE PRESSURE-TEMPERATURE RELATIONS OF SATURATED STEAM

By Lionel S. Marks, Cambridge, Mass.
Member of the Society

The relation between the pressure and temperature of saturated steam is known with great accuracy for the range of temperatures from 32 deg. to 400 deg. fahr., or for the approximately corresponding pressure range from 0.1 lb. to 250 lb. per sq. in. abs. Within that range the experimental values of Regnault and of other investigators agree very closely with the recent and authoritative work of Scheel and Heuse[1] and of Holborn and Henning.[2] It is probable that

TABLE 1 PRESSURE OF SATURATED STEAM AT HIGH TEMPERATURES

Temperature, Deg. Fahr.	Pressures in Atmosphere according to		
	Battelli	Cailletet and Colardeau	Knipp
400	16.8	16.8	.....
500	48.6	46.6	47.1
600	109.7	107.1	112.5
680	186.7	189.0	207.7

the temperature corresponding to any pressure within that range is now known accurately to within $\frac{1}{20}$ deg. fahr.

2 At higher pressures and temperatures there is no such agreement between the results of different investigations. Until the past year the most important work in this part of the field had been done by Battelli, by Cailletet and Colardeau and by Knipp. Table 1

[1]Annalen der Physik, 1910, vol. 31, pp. 715-735. The range covered was from 32 deg. to 122 deg. fahr.

[2]Annalen der Physik, 1908, vol. 26, pp. 833-883. The range covered was from 122 deg. to 400 deg. fahr.

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

(from Marks and Davis' Steam Tables, p. 95) shows the very considerable divergence between the results obtained by these three investigations, particularly at the higher temperatures.

3 During the past year there have been published[1] the results of an investigation carried out at the Reichsanstalt by Holborn and Baumann on the pressure-temperature relations of saturated steam from 400 deg. fahr. to the critical temperature. Considerable weight must certainly be given to any investigation coming from that source and in this particular case, all the internal evidence would indicate that more weight should be given to these results than to those of any of the previous investigators. Of the earlier investigations, the internal evidence clearly indicated that the work of Cailletet and Colardeau was probably the most accurate and for that reason their results have been adopted in recent steam tables. The new values by Holborn and Baumann differ markedly from the values obtained from the three earlier investigations, but are much closer to the values of Cailletet and Colardeau than to the others. It seems worth while to describe briefly the methods and results of this new work.

4 The measurement of vapor pressure may be either by the statical or the dynamical method. In the statical method the liquid and its vapor are maintained at constant temperature and the corresponding pressure is measured. The simplest example of the statical method is that used by Regnault in his experiments with low-pressure steam. For this work the liquid was inserted above the mercury in a barometer tube, the upper part of which was kept in a constant temperature bath; the depression of the mercury column measured the vapor pressure corresponding to the temperature of the bath. In the dynamical method the pressure is kept constant and the corresponding temperature measured. The pressure is maintained by air or other gas acting on top of the liquid, the liquid is heated continuously, and the vapor which forms is condensed continuously and is returned by gravity. The process is similar to that customarily employed for the determination of boiling points at atmospheric pressure, an artificial atmosphere of any desired pressure being maintained over the liquid.

5 The work of Holborn and Henning on saturation pressures from 120 to 400 deg. fahr. was by the dynamical method. This new work by Holborn and Baumann is by the statical method.

[1]Annalen der Physik, 1910, vol. 31, pp. 945–970.

The water was contained in a steel vessel surrounded by a constant temperature bath; absolute measurements of the pressure were obtained by means of a weighted rotating plunger; the volume of water in the vessel could be varied either continuously at an approximately uniform rate, or intermittently.

6 In presenting their results, Holborn and Henning used as a standard of reference the following formula of Thiesen:

$$(t+273)\ \log \frac{p}{760} = 5.409\ (t-100)\ - 0.508 \times 10^{-8}\ [(365-t)^4 - 265^4]$$

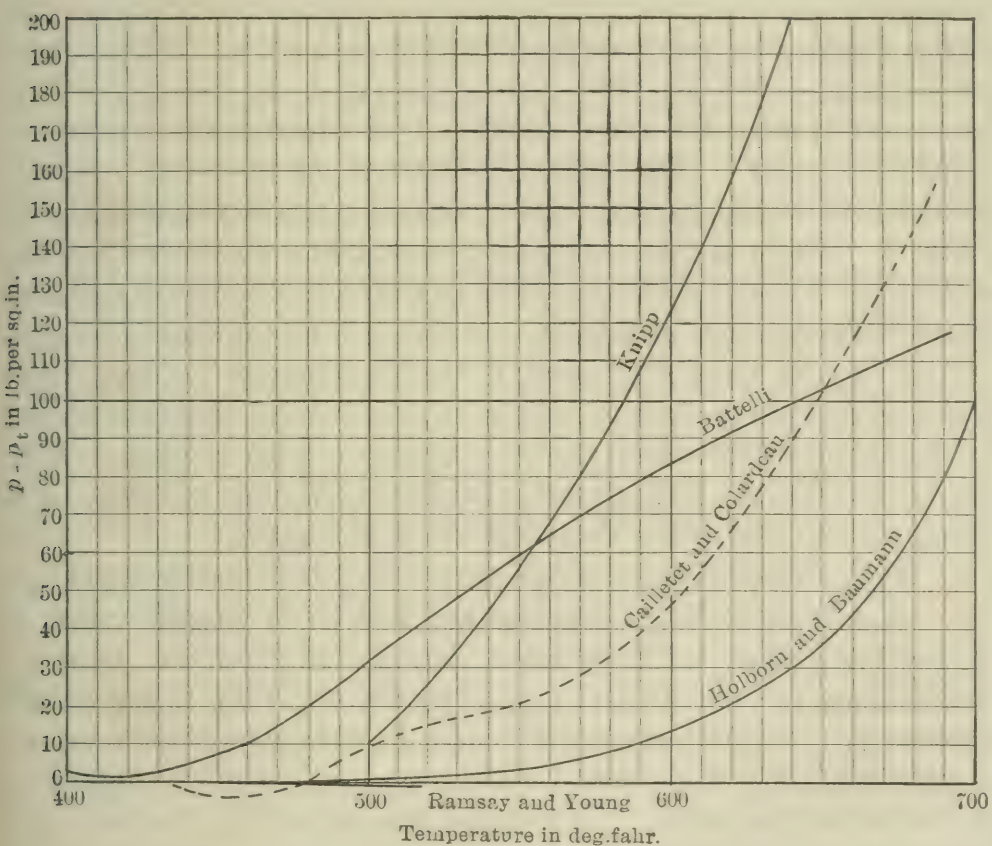


FIG. 1 CURVES SHOWING DEVIATION OF PRESSURE p OBSERVED BY VARIOUS INVESTIGATORS FROM PRESSURE p_t CALCULATED FROM MODIFIED THIESEN EQUATION FOR TEMPERATURE ABOVE 400 DEG. FAHR.

in which the pressure p is in mm. of mercury at 0 deg. cent. and the temperature t is in deg. cent. The deviation of their experimental values of the temperature from those calculated from the formula is small and is given by a correction curve.[1] The Thiesen formula also fits the Scheel and Heuse results up to 120 deg. fahr. with great

[1]Marks and Davis, Steam Tables, p. 94.

exactness. Holborn and Baumann have followed a similar procedure in presenting their results for temperatures above 400 deg. fahr. The equation they have adopted as a standard of reference is a modification of the Thiesen equation. Their equation is

$$(t+273)\log\frac{p}{1.0333} = 5.3867\ (t-100) - 0.5262\times 10^{-8}\ [(365-t)^4-265^4]$$

where p is in kg. per sq. cm. and t is in deg. cent.

7 This new equation fits the experimental data extremely well from 212 deg. to 400 deg. fahr. The maximum difference between the observed and calculated pressures at any temperature within that range is 2 mm. of mercury or about 0.04 lb. per sq. in. At temperatures in excess of 500 deg. fahr. this equation does not give good results and its error increases very rapidly as the temperature increases beyond that value. The difference between the observed pressures and those calculated from the above equation are shown graphically in Fig. 1. There is also added for comparison the corresponding deviations of the experimental values of Battelli, of Cailletet and Colardeau, of Knipp, and of Ramsay and Young from the same equation. It is obvious at once how great is the discrepancy between these various experimental results, and it is also evident that the results of Cailletet and Colardeau agree much better with the latest work than do the others. The agreement with the results of Ramsay and Young, which extend only to about 520 deg. is good.

8 Holborn and Baumann have presented their results in tabular form. A column in Table 3 gives their results in English units, as deduced from their table by interpolation. It gives the pressures corresponding to each 10 deg. fahr. from 400 deg. fahr. to the critical temperature. As the second differences in the original table do not run smoothly, it will be found that the same is true of the second differences in this table. The values determined by Cailletet and Colardeau are included in the same table for purposes of comparison.

9 There is but little doubt that these new values should be accepted in preference to all earlier values. It is probable that they are not final; the investigators noted, contrary to the observations of earlier investigators, that the water acted on the walls of the steel vessel and that after repeated heatings over 570 deg. fahr. a small quantity of iron went into solution in the water and that the water became colored on standing in the air. There was evidence also that gas was formed in the vessel as a result of this action. It is possible that the same actions have taken place with the earlier investigations in which steel vessels were used, but that they passed unnoticed.

Notwithstanding this evidence of the presence of small quantities of impurities in the vessel at the highest temperatures, it seems probable that the results have a high order of accuracy. The probable degree of accuracy is so high that it is doubtful indeed whether

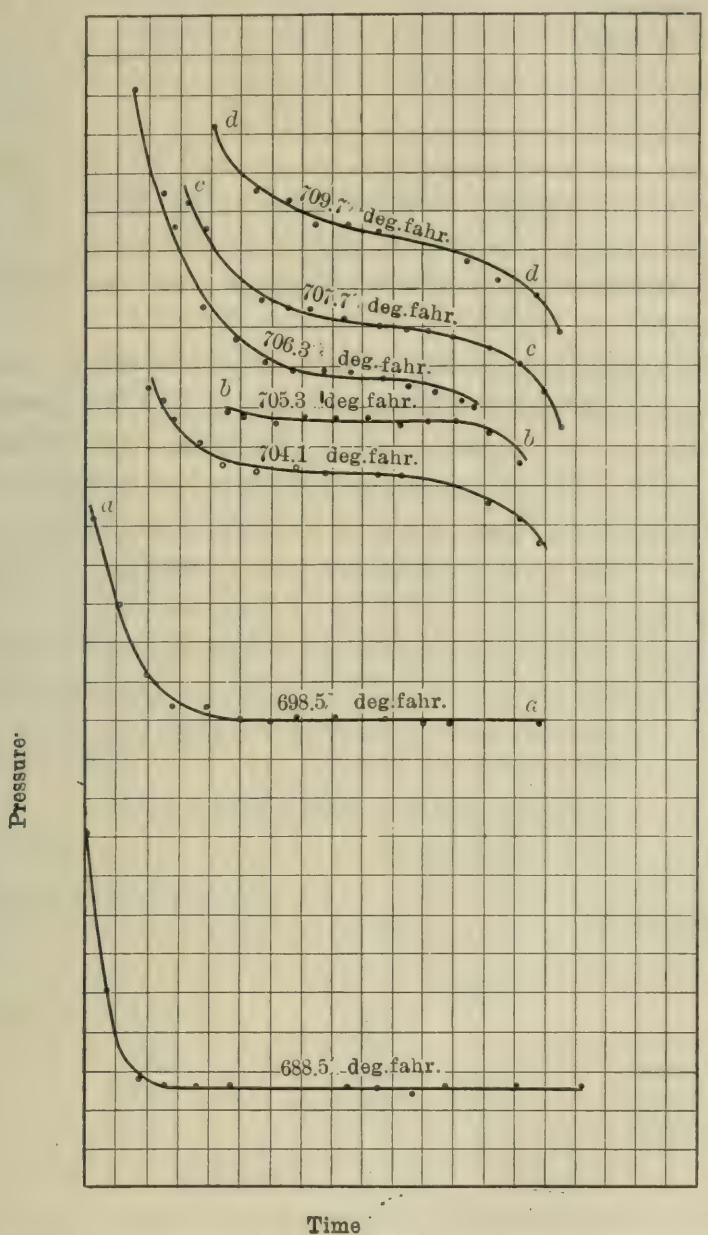


FIG. 2 ISOTHERMALS FOR STEAM—HOLBORN AND BAUMANN

any other investigator will be tempted to explore this field for a long time.

10 It has been previously noted that there was in this investigation the possibility of varying continuously and at an approximately

constant rate, the amount of water contained in the steel vessel. If the vessel is initially full of water at a pressure in excess of the saturation pressure corresponding to the constant temperature at which the vessel is being kept, the gradual outflow of water will result in a fall of pressure until the saturation pressure is reached. Evaporation will then take place and the pressure will remain steady until all the water has escaped. If the pressures are observed and are plotted against time, the observations will give curves such as *aa*, *bb*, Fig. 2. These are isothermal curves and will be true pressure-volume curves if the rate of outflow of the water is kept constant. If the rate of outflow is not constant, as in these investigations, the curves will be altered in shape, but nevertheless have a horizontal part.

11 If the vessel is maintained at a temperature in excess of the critical temperature and the fluid contents of the steel vessel are permitted to flow out at a constant rate, the pressure curve will be such as *cc* or *dd*, and will not have any constant pressure, or horizontal section, since there cannot be any evaporation. The variation of pressure with time was investigated for a number of different isothermals in the vicinity of the critical temperature with the results shown in Fig. 2. They indicate that the critical temperature of water is about 374.6 deg. cent. or 706.3 deg. fahr. and the corresponding pressure 225 kg. per sq. cm. or 3200 lb. per sq. in.

TABLE 2 CRITICAL DATA FOR WATER

Investigator	Date	Critical Temperature, Deg. Fahr.	Critical Pressure, Lb. per Sq. In.
Nadejdine	1885	676.6	....
Battelli	1890	687.7	2859
Cailletet and Colardeau	1891	689	2944
Traube and Teichner	1904	705.2	....
Holborn and Baumann	1910	706.3	3200

12 The critical constants as determined in this way differ from the earlier values. The more important of these are given in Table 2.

13 It will be seen that the critical values have been continuously raised by the later investigators. The practical agreement of the last two determinations of the critical temperature is probably indicative of a fairly correct value. The method of Holborn and Baumann is well adapted to give good results.

14 Accepting the data of Holborn and Henning and of Holborn and Baumann as the best available, and as probably of very considerable accuracy, it becomes interesting to ascertain whether any formula can be found to represent them accurately. Most of the formulae which have been suggested become quite inaccurate above 400 deg. fahr. Prof. Robert H. Smith[1] has suggested the formula

$$p = -0.16 + 0.0075\,t + 2.816 \times 10^{-10}\,t^{4.588}$$

which fits the Cailletet and Colardeau values extremely well up to 600 deg. fahr. with a maximum error of 0.3 lb. per sq. in. at 300 deg. fahr., but with large errors above 600 deg. fahr. The writer has not tried an equation of that general form with the new values; but there is little doubt that a satisfactory equation of that form could be found. It has seemed better, however, to proceed along different lines in looking for the desired equation.

15 The pressure and temperature of a substance can be expressed in terms of any unit. In seeking for a form of statement of the pressure-temperature relations of a saturated vapor, which may possibly apply to the vapors of many, or even of all substances, it is desirable to express the pressures and temperatures *not* as multiples of the usual units (lb. sq. per in. and deg. fahr.), but as fractions of the absolute critical pressure and temperature of the substance under consideration. Taking 3200 lb. per sq. in. and 1106 deg. fahr. as the critical constants for water, a pressure of 1600 lb. per sq. in. becomes 0.5 in the new units, and an absolute temperature of 553 deg. has the same value in the new units. Pressures and temperatures expressed in these units are called reduced pressures and temperatures, and substances at the same reduced pressures and temperatures are said to be in corresponding states.

16 There has been considerable speculation as to the relations between the properties of substances which are in corresponding states. The most obvious hypothesis, suggested by a cursory examination of the saturation data for various substances, is that substances at the same reduced temperatures have the same reduced pressures. For example, water at a reduced temperature of 0.7 (=816.4 deg. abs.) has a pressure of 146 lb. per sq. in. or a reduced pressure of 0.0456. Ammonia (critical temperature 730 deg. abs., pressure 1690 lb. per sq. in.) at a reduced temperature of 0.7 (511 deg. abs.) should have according to this hypothesis, the same reduced pressure (0.0456) or

[1]The Engineer (London), August 26, 1910.

a pressure of 77.1 lb. per sq. in. The observed pressure is, however, 90.7 lb. per sq. in. The hypothesis will be found to give only a very rough approximation to the observed facts in many cases. If the hypothesis were true we should have the equation

$$\frac{p}{p_c} = f\left(\frac{T}{T_c}\right)$$

where p_c and T_c are the critical constants and p is the absolute saturation pressure corresponding to any absolute temperature T.

17 The equation given in Par. 16 is assumed by van der Waals[1] to have the special form

$$\log \frac{p_c}{p} = a\left(\frac{T_c}{T} - 1\right)$$

where a is a constant with the same value for all substances, with an approximate value of 3. This equation gives results which are near enough to the facts to suggest that it has some logical basis, but which gives errors too great for practical use. It has the positive merit of being accurate at the critical temperature, but it is found that the value of the constant a differs with the substance. Nernst[2] gives the following values of a which, he says, hold for the range $\frac{T_c}{T} = 1.5$ to 2.0, a range which is the important range for water.

Hydrogen	1.83
Argon	2.17
Nitrogen	2.45
Oxygen	2.53
Carbon bi-sulphide	2.63
Chloroform	3.00
Ether	3.15
Propylacetate	3.39
Ethyl alcohol	4.05

He gives also a series of curves, Fig. 3, which seem to show a straight line relation between $\log \frac{p_c}{p}$ and $\left(\frac{T_c}{T} - 1\right)$.

18 If the van der Waals equation were true for all these substances, the lines would all be straight lines; and if a were constant they would be superposed. Applying the equation to saturated steam and finding the values of a corresponding to different reduced

[1]Continuität der gasförmigen und flüssigen Zustandes, p. 147.
[2]Gottingische Nachrichten, Math.-Phys. Klasse 1906.

temperatures, it becomes evident at once that a is not a constant, but varies as in the curve, Fig. 4. This curve goes through a minimum for a value of $\frac{T}{T_c}$ approximately of 0.78. The curve suggests that a varies with $\frac{T_c}{T}$ according to the equation

$$a = b\left[1 + c\frac{T}{T_c} + d\left(\frac{T}{T_c} - 0.78\right)^2\right]$$

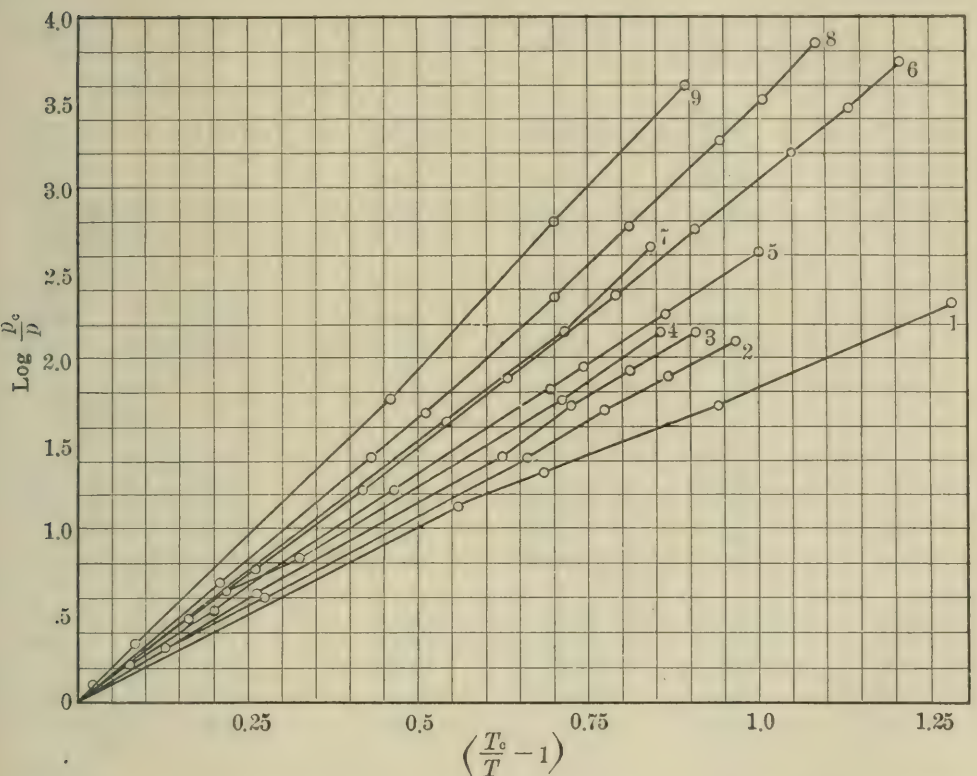


FIG. 3 CURVES SHOWING APPLICABILITY OF VAN DER WAALS EQUATION TO (1) HYDROGEN, (2) ARGON, (3) KRYPTON, (4) OXYGEN, (5) CARBON BI-SULPHIDE, (6) FLUORBENZOL, (7) ETHER, (8) PROPYLACETATE, (9) ETHYL ALCOHOL, TAKEN FROM NERNST

where b, c and d are constants. By a few trials it was found that 0.7875 gave somewhat better results than the approximate value of 0.78 in the last term. The van der Waals equation as modified for the variation of a becomes

$$\log\frac{p_c}{p} = b\left[1 + c\frac{T}{T_c} + d\left(\frac{T}{T_c} - 0.7875\right)^2\right]\left(\frac{T_c}{T} - 1\right)$$

19 This equation is very sensitive to slight variations in the assumed values of the critical constants. Assuming the critical pres-

sure to be 3200 lb. per sq. in., the corresponding critical temperature which was found to give calculated results in closest accord with the experimental results was 706.1 deg. fahr. A variation of $\frac{1}{10}$ deg.

TABLE 3 EXPERIMENTAL AND CALCULATED PRESSURES OF SATURATED STEAM FROM 400 DEG. FAHR. TO THE CRITICAL TEMPERATURE

Temperatures, Deg. Fahr.	Pressures, Lb. per Sq. In.			Deviation of Formula From Holborn and Baumann	
	Cailletet and Colardeau	Holborn and Baumann	By Formula	Lb. per Sq. In.	Percentage
400	247.1	246.99	247.10	+0.11	0.044
410	276.4	276.34	276.47	+0.13	0.047
420	308.4	308.33	308.47	+0.14	0.045
430	343.2	343.18	343.26	+0.08	0.023
440	380.8	380.92	381.02	+0.10	0.026
450	421	421.85	421.87	+0.02	0.0047
460	465	465.95	466.04	+0.09	0.019
470	573	513.65	513.66	+0.01	0.0019
480	565	565.08	564.93	−0.15	−0.026
490	622	620.18	620.05	−0.13	−0.021
500	684	679.26	679.18	−0.08	−0.012
510	751	742.55	742.56	+0.01	0.0013
520	822	810.31	810.37	+0.06	0.0074
530	897	882.58	882.82	+0.24	0.027
540	977	959.85	960.15	+0.30	0.031
540	1062	1042.2	1042.6	+0.4	0.038
560	1152	1130.2	1130.3	+0.1	0.0089
570	1247	1223.7	1223.7	0	0
580	1349	1323.0	1322.9	−0.1	−0.0076
590	1458	1428.3	1428.1	−0.2	−0.014
600	1574	1539.9	1539.8	−0.1	−0.0065
610	1697	1657.8	1658.1	+0.3	0.018
620	1827	1782.9	1783.3	+0.4	0.022
630	1965	1915.3	1915.9	+0.6	0.031
640	2111	2055.1	2056.0	+0.9	0.044
650	2265	2203.1	2204.2	+1.1	0.049
660	2428	2359.2	2360.5	+1.3	0.055
670	2599	2523.4	2525.6	+2.2	0.067
680		2697.1	2699.7	+2.6	0.096
690		2882.3	2883.3	+1.0	0.035
700		3080.4	3076.8	−3.6	−0.117
706.1		3200.0	3200.0	0	0

fahr. in the assumed critical temperature has a very marked effect on the calculated pressures at the highest temperatures. The value of the critical temperature given by Holborn and Baumann is 706.3

deg. fahr., but it is certain that their method cannot give results which are reliable to within a few tenths of a degree. It is probable that an equation which fits the experimental facts for a range of over 600 deg. fahr. may be extrapolated with some certainty for 20 deg. or 30 deg. In obtaining the constants in the equation it was consequently assumed that the critical temperature was 706.1 deg. fahr. or

$$T = 706.1 \text{ deg.} + 459.64 \text{ deg. fahr.}$$

After numerous trials the following equation was finally adopted

$$\log \frac{p_c}{p} = 3.006854 \left(\frac{T_c}{T} - 1 \right) \left[1 + 0.0505476 \frac{T}{T_c} + 0.629547 \left(\frac{T}{T_c} - 0.7875 \right)^2 \right]$$

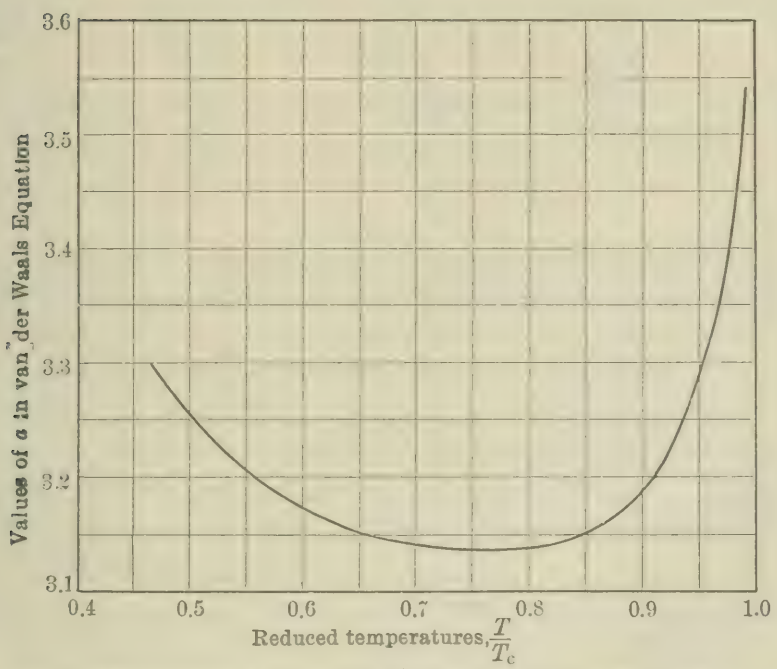


FIG. 4 CURVE SHOWING VARIATION OF a IN VAN DER WAALS' EQUATION FOR SATURATED STEAM

Inserting the chosen values of the critical constants this reduces to

$$\log p = 10.515354 - 4873.71\ T^{-1} - 0.00405096\ T + 0.000001392964\ T^2$$

20 The agreement of the pressures calculated from this equation with the experimental results of Holborn and Baumann is very striking and is shown in Table 3, which gives the calculated pressures and the differences from the experimental results, expressed both in lb.

per sq. in. and in percentages. It will be seen that from 400 deg. to 650 deg. the difference is nowhere so great as $\frac{1}{20}$ of 1 per cent. From

TABLE 4 EXPERIMENTAL AND CALCULATED PRESSURES OF SATURATED STEAM FROM 32 DEG. TO 400 DEG. FAHR.

Temperatures, Deg. Fahr.	Pressure, Lb. per Sq. In.		Deviations of Formula From Tabulated Values	
	Marks and Davis' Steam Tables	By Formula	Lb. per Sq. In.	Percentage
32	0.0886	0.088563	−0.000037	−0.042
50	0.1780	0.17765	−0.00035	−0.196
100	0.946	0.946	0	0
150	3.714	3.707	−0.007	−0.188
200	11.52	11.504	−0.016	−0.139
250	29.82	29.802	−0.018	−0.060
300	67.00	67.00	0	0
350	134.6	134.60	0	0
400	247.1	247.10	0	0

650 deg. to the critical temperature the maximum difference is slightly greater than $\frac{1}{10}$ of 1 per cent but becomes zero at the critical

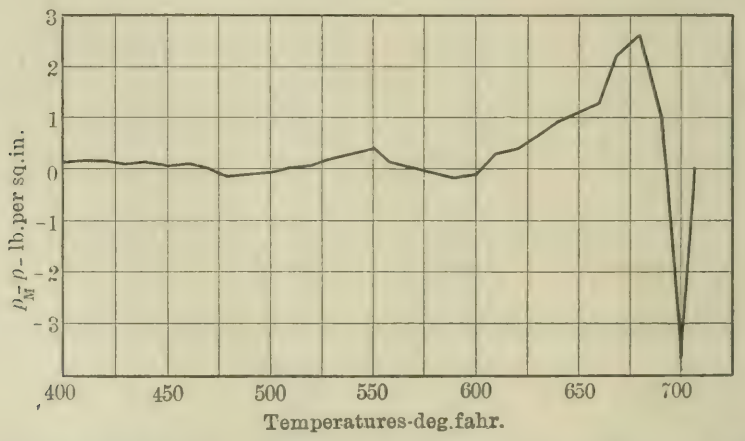


FIG. 5 CURVE SHOWING DIFFERENCE BETWEEN CALCULATED PRESSURE p_m AND OBSERVED PRESSURE p FOR TEMPERATURE RANGE FROM 400 DEG. FAHR. TO THE CRITICAL TEMPERATURE

temperature. The curve, Fig. 5, shows the differences in lb. per sq. in. It is probable that but little weight should be given to the differences in the neighborhood of the critical temperature in view of the decreasing accuracy of the experimental results in that vicinity.

21 Below 400 deg. fahr. the agreement of the equation with the experimental results is shown in Table 4. The experimental results are those of Holborn and Henning from 120 deg. to 400 deg., and of Regnault and other investigators below 120 deg. The differences are very small as expressed in lb. per sq. in., but amount nearly to $\frac{1}{6}$ of 1 per cent in some cases. These differences are not greater than the variations among the best experimental values in this part of the temperature range. For example, the new and excellent work of Scheel and Heuse gives a pressure of 0.088444 lb. at 32 deg. fahr. as against 0.0886 lb. given in Table 4 (from Regnault's work) and 0.088563 lb. calculated from the equation.

22 The equation given in Par. 19 has, it is believed, the advantage over previous equations of greater simplicity, greater accuracy, and also of the great ease in finding $\frac{dp}{dt}$ for use in the Clapeyron equation.

23 I am indebted to H. J. Macintire, Jun.Am.Soc.M.E., for most of the numerical work in obtaining the equations.

DISCUSSION

W. D. Ennis. If in Fig. 2 the lengths of the approximately horizontal portions of the curves are measured, the following is obtained: At 704.1 deg. fahr., $1\frac{1}{2}$ squares; at 705.3 deg. fahr., 3 squares; at 706.3 deg. fabr., $\frac{1}{2}$ square. Since the critical temperature is that at which the horizontal portion becomes zero, the inference is justifiable that that temperature is not far from 706 deg. The 706.1 deg. assumed in the final formula in Par. 19 has a sufficient basis, and the old value, 689 deg., is certainly too low.

Professor Marks may perhaps be able to state from his examination of the Holborn and Baumann research, whether the abscissae in Fig. 2 are volumes as well as times. If so, the lengths of the horizontal portions of the curves should steadily decrease as the temperature increases. The measurements given indicate that they do not steadily decrease.

The final equation for pressure-temperature is in a particularly useful form. Numerical values for the derivative $\frac{dp}{dt}$ may be obtained therefrom with a minimum of computation. In this respect the new formula is far superior to that of Thiesen or to the old equations of the form

$$\log p = a - bd^n - c\theta^n$$

where n is a function of the temperature, or even to the Rankine equation in which log p is a function of the -1 and -2 powers of the temperature. The accuracy of the equation surpasses its convenience, and within the most extreme range of power engineering the agreement with recent tabular values may be regarded as exact.

R. C. H. Heck. Two years ago the writer compared various data as to the pressure of steam at high temperatures, and traced a curve which seemed to show the most probable trend of the true relation. On the appearance of the Holborn-Baumann determination a year later, the assumed curve was found to be well justified up to 1000 lb. pressure, or to 550 deg. fahr., which is about as far as there is any real use in attempting to extend the steam table. The whole comparison, Fig. 6, seems worthy of a place in connection with Professor Marks' paper.

The only satisfactory way to handle such a discussion is to take a mathematical formula, the simpler the better, which follows the general trend of the experimental values, and plot the small departures from that equation. Following Henning,[1] the formula of Thiesen was used, with some modification. It is preferably written in the general form

$$\log \frac{p}{p_a} = A \frac{t - t_a}{T} - B \frac{(t_c - t)^4 - (t_c - t_a)^4}{T} \quad \ldots\ldots\ldots [1]$$

in which p_a is the pressure of the atmosphere, t_a the corresponding temperature, t_c the critical temperature, taken as 365 deg. cent., or 689 deg. fahr., and T is absolute temperature at t. The coefficient A is the same with either thermometric scale, but the centigrade value of B must be divided by 1.8^3 when changing to fahrenheit temperatures.

The original Thiesen coefficients were

$$A = 5.3807,\ B = 0.508 \times 10^{-8}\ (0.87106 \times 10^{-9}) \quad \ldots\ldots\ldots [2]$$

The writer found that by changing these to

$$A = 5.3807,\ B = 0.540 \times 10^{-8}\ (0.92593 \times 10^{-9}) \quad \ldots\ldots\ldots [3]$$

an almost perfect agreement with the Holborn-Henning[2] determination from 100 to 200 deg. cent. was secured, as appears in Table 5.

[1] Annalen der Physik, 1907, vol. 22, pp. 609–630.

[2] Annalen der Physik, 1908, vol. 26, pp. 833–883.

In what follows, equation [1] with these constants will be called Formula *A*.

As noted in the paper, Holborn and Baumann[1] used the values

$$A = 5.3867,\ B = 0.5262 \times 10^{-8} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

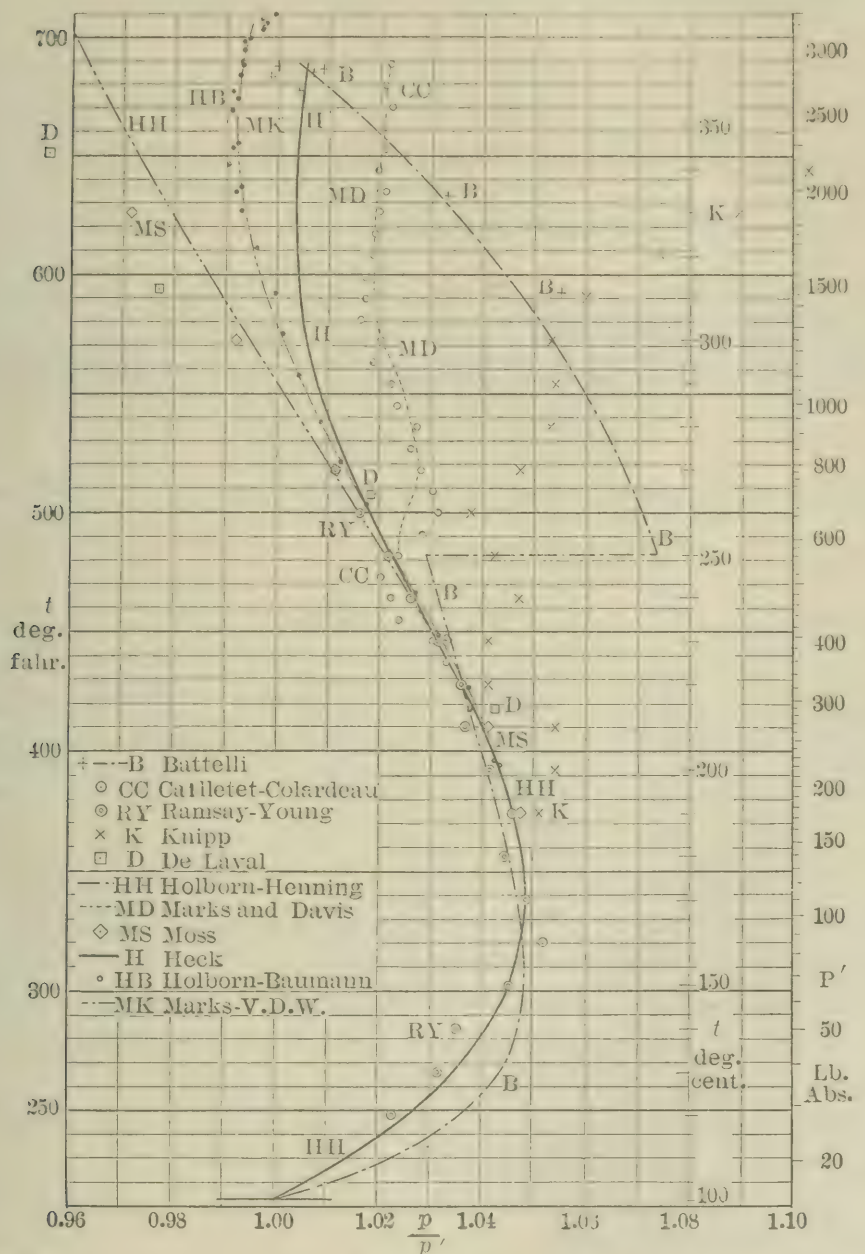


FIG. 6 COMPARISON OF DATA RELATING TO THE PRESSURE OF STEAM AT HIGH TEMPERATURES

and obtained excellent agreement with the experiments below 200 deg.

[1] Annalen der Physik, 1910, vol. 31, p. 945

As a foundation for the comparison in Fig. 6, the first or principal term of the second member of Thiesen's equation was used to calculate a reference pressure p'. The coefficient A was changed, by trial, with the purpose of keeping the range of data near to the reference line, and was fixed at 5.52. From the formula

$$\log \frac{p'}{p_a} = 5.52 \frac{t - t_a}{T} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

were computed values of p'. Then at any temperature t and pressure p, the ratio $\frac{p}{p'}$ is plotted to the scale at the bottom of Fig. 6; or, in effect, the difference, $p - p'$, is laid off as a fraction or percentage of p', the latter just reaching to, or being represented by, the vertical line AA. Thus to piot differences as percentages makes the

TABLE 5 TRIAL OF FORMULA *A*

Pressure p, in Mm. of Mercury; Column *HH*, from Holborn-Henning Table; Column Form. by Formula

t Deg. Cent.	p *HH*	p Form.	t Deg. Cent.	p *HH*	p Form.
110	1074.5	1074.53	160	4633	4633.2
120	1488.9	1489.00	170	5937	5936.4
130	2025.6	2025.71	180	7514	7513.8
140	2709.5	2709.67	190	9404	9404.1
150	3568.7	3568.77	200	11647	11647.2

scheme of comparison equally sensitive over the whole range of temperature. The scale of p' will give an approximate idea of the variation of pressure with temperature, obviating the need of referring to some table when examining the diagram. Naturally, Formula *A* would have given a preferable reference line, but the work was largely completed before that formula was established.

The various data are named on Fig. 6 in chronological order, and may be briefly described as follows:

Battelli[1] fixed two sets of constants for the Biot formula (the one which Regnault preferred)

$$\log p = a + b\alpha^t + c\beta^t \quad \ldots\ldots\ldots\ldots\ldots\ldots [6]$$

These change at 250 deg. cent., but the plot shows that the two sec-

[1]Memorie d. reale Accad. d. Scienze di Torino, 1891, vol. 41; and 1893, vol. 43.

tions decidedly fail of continuity. With the upper section are given the experimental points on which it was based.

The Cailletet-Colardeau[1] measurements, from 225 deg. cent. upward, show a sudden jog of about 1.5 per cent near 250 deg., then proceed very regularly. With these points is traced a broken line from the high-temperature end of the Marks and Davis tables.

The Ramsay-Young[2] experiments run only to 270 deg. cent., but are entirely consistent. In the fixing of the writer's curve they were given predominant weight, as against any tendency to make the curve rise to the Cailletet-Colardeau plot.

The experiments of Knipp[3] do not deserve a place among data worthy of serious consideration.

A few observations reported from the original De Laval turbine works at Stockholm[4] run below Formula A, or the extended HH curve. With these are given some points computed from the Roche formula (Regnault's K), recommended by Moss[5]

$$\log p - A^{\left(B - \frac{C}{T}\right)^{1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [7]$$

where A, B and C are constants and T is absolute temperature. This evidently runs low.

The writer's extended curve H was based upon the modified Thiesen equation, Formula A. The latter gives the dotted curve marked HH. The character of the deflection was determined by adopting a smooth and gradually increasing departure of the derivative $\frac{dp}{dt}$ from the range of values computed by Formula A, then working back to the primitive. The general idea was to parallel the trend of the CC points, and to aim at the middle of the group of B points near the critical temperature.

The Holborn-Baumann experiments are plotted as points, and with them is drawn the curve of Marks' adapted van der Waals equation, from Table 3, column 4. Of course, this last determination supercedes all others, as a statement of scientific fact. The writer, having made out a smooth and consistent extension of the steam table up

[1] Annales de Chimie et de Physique, 1892, series 6, vol. 25, pp. 519–534.
[2] Phil. Trans. Roy. Soc., 1892, vol. 193A, pp. 107–130.
[3] Physical Review, 1900, vol. 11, pp. 129–154.
[4] Engineering, 1907, vol. 83, p. 1.
[5] Physical Review, 1908, vol. 26, pp. 439–447.

to 550 deg. fahr., is gratified to find that the absolute error at this upper limit of usefulness does not exceed 0.25 per cent. A difference of 1.25 per cent at the critical temperature and the raising of the latter by 17 deg. fahr. can be viewed with equanimity.

It is of interest to note that Holborn and Baumann measured pressures with a weighted plunger. The earlier experiments were made with a closed mercury manometer; at very high pressures, the compression of the air above the mercury column becomes very much the major component, and the difficulty of precise determination increases rapidly.

THE AUTHOR. The author is glad that Professors Ennis and Heck concur in his opinion that the pressure-temperature relations of saturated steam may now be regarded as practically settled for all purposes of the engineer by the investigations reported in this paper. It is now possible to eliminate from the steam tables one of the most considerable of the hitherto existing uncertainties.

In reply to the question of Professor Ennis, the abscissae in Fig. 2 do not accurately represent volumes. They represent times, and the times depend primarily upon the rate of leakage past the weighted plunger which was used for measuring pressures. It is not possible to base any quantitative deductions upon measurements of the abscissae of Fig. 2.

It is interesting to see that the conclusions to which Professor Heck has come, as a result of his careful examination of the earlier investigations, are in such close accord with the latest investigations, and that they are a demonstration of the value of such analysis.

No. 1319

PRESSURE-RECORDING INDICATOR FOR PUNCHING MACHINERY

By Prof. Gardner C. Anthony, Tufts College, Mass.

Member of the Society

The design of this device was assigned as the subject for a mechanical engineering thesis at Tufts College, in the endeavor to determine such values as the maximum pressures for which punching machines should be designed, the point of maximum stress in the punching of plates, the advantages to be derived from the use of shearing punches, and the effect of increased clearance between punch and die on the maximum stress and the effect of time on the flow of metal in punching.

2 The apparatus was designed to enable indicator diagrams to be taken under the conditions of actual shop practice by applying the mechanism to punching machinery now in use. It was first used in May 1909, at the New England Iron Works, South Boston, where it was applied to a Bisbee-Endicott lever-type punch, for which it was designed. Tests were made on plates of $\frac{1}{4}$ in., $\frac{7}{16}$ in. and $\frac{1}{2}$ in. thickness, with punches of $\frac{1}{2}$ in., $\frac{5}{8}$ in. and $\frac{3}{4}$ in. diameter.

3 Only such tests were made at that time as were deemed necessary to demonstrate the efficiency of the apparatus, since before applying it to a series of tests it was thought desirable to compare the indicator diagrams already taken with those which might be obtained from its application to a testing machine, as illustrated in Fig. 1. This was done in June 1910, when it was applied to a 60,000-lb. Olsen testing machine. The results demonstrated the desirability of continuing the experiments on this machine, since there was but a slight variation in the cards, due to the increased time of punching, while a much greater range of speeds was possible.

4 The number of tests now made are insufficient for conclusive

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

evidence concerning the problems relating to shearing stresses, but it was thought desirable to present this application of an indicator as a useful addition to recording devices for testing machines, and to give some results of tests already made for the purpose of stimulating experimental work in this direction.

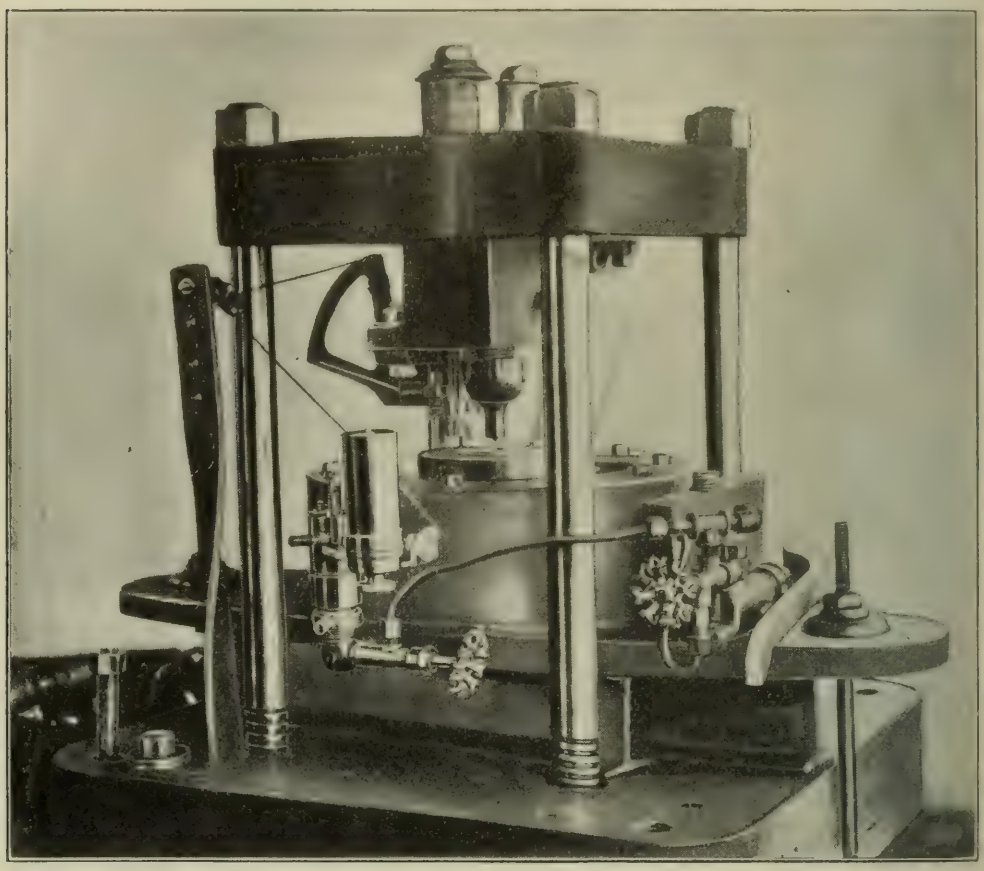


FIG. 1 PRESSURE-RECORDING APPARATUS APPLIED TO TESTING MACHINE

5 The principle employed in the design of the pressure-recording mechanism was that of the Emery testing machine, save that the hydraulic chamber communicates directly with an ordnance indicator, as in Fig. 2, which shows a cross-section of the compression cylinder with the indicator attached and the motion mechanism bolted to the crosshead of a punching machine. The die N is clamped to a piston P, which rests on a thin diaphragm D enclosing a small volume of oil, the pressure of which is recorded by the indicator. The piston has an area of 50 sq. in. and is guided by two annular steel disks fixed between

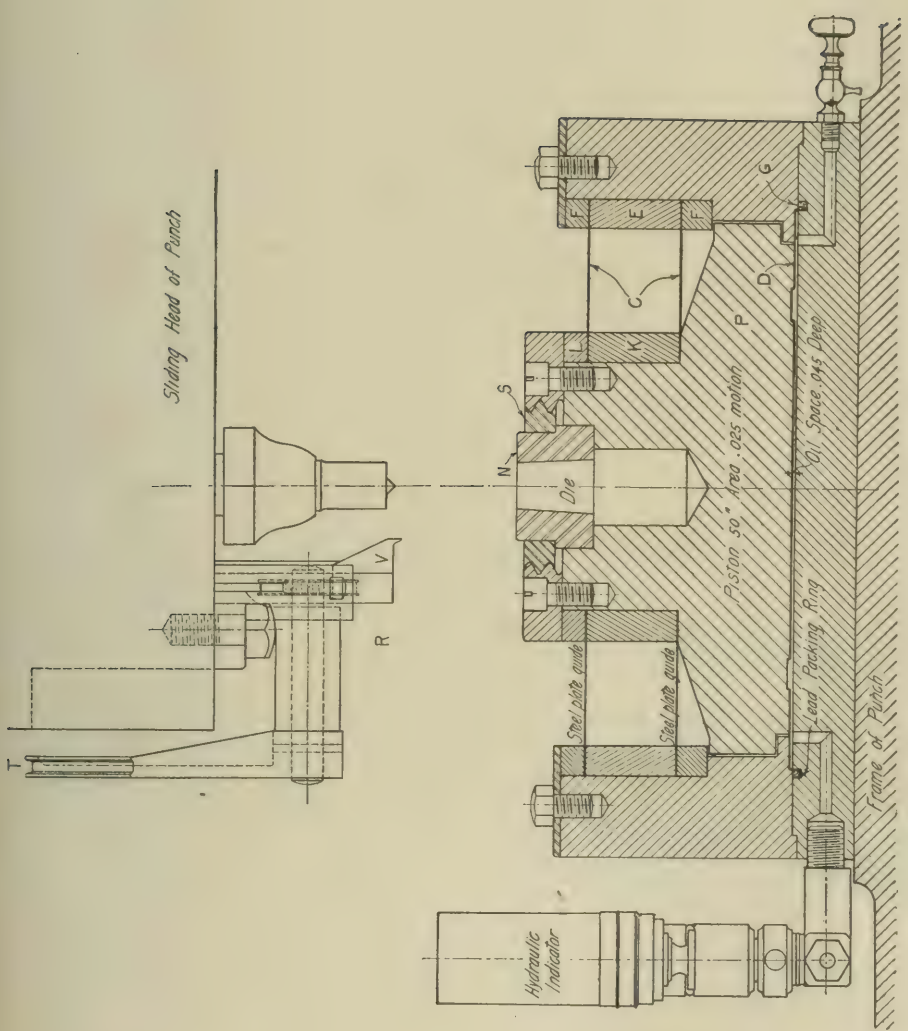


FIG. 2 SECTION THROUGH COMPRESSION CYLINDER

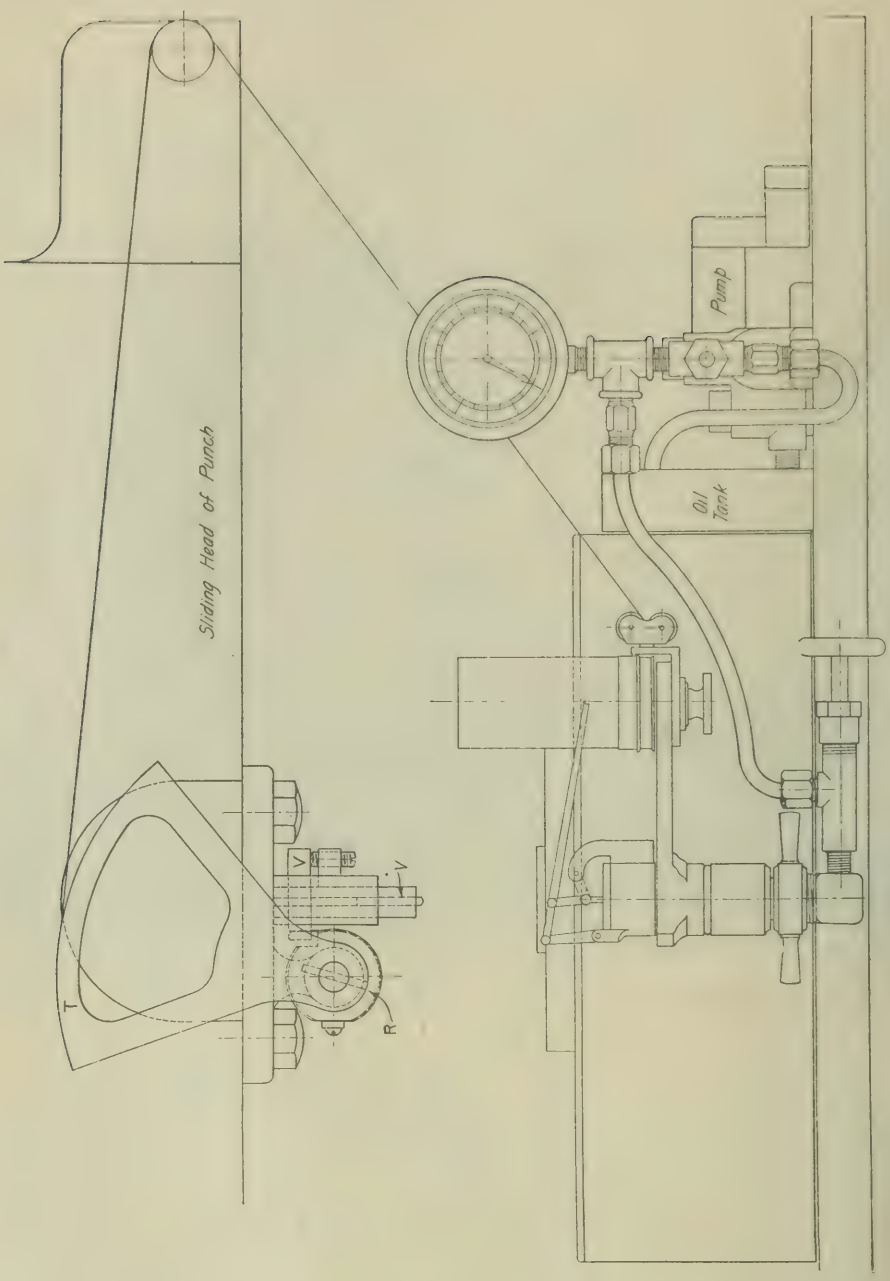


FIG. 3 DIAGRAM OF RECORDING APPARATUS

rings E and F and K and L. These serve to centralize the piston and to eliminate all frictional losses due to surface contact. The die N is held in place by an interrupted screw S, by means of which the die can be removed quickly and the burs, or punchings, removed from the space below. The diaphragm is made of sheet brass 0.007 in. thick and fitted to the grooved space G, at which place the pressure chamber is made tight by a lead packing ring. An air valve and an indicator pipe communicate with this space. The recess on the lower face of the piston serves to make the diaphragm taut by being forced into place when first under pressure, and also permanently to center the piston. The piston and die have a possible motion of 0.025 in., the limits being determined by the lower ring F and the inner flange at the bottom of the cylinder. Only a small part of this motion is

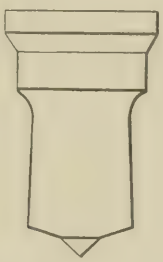


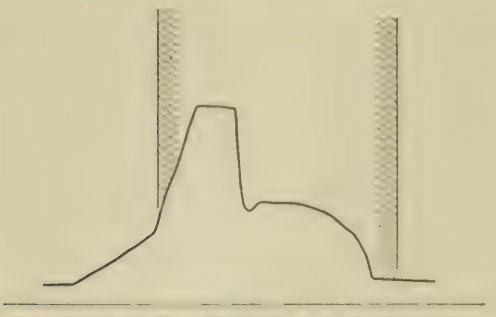
FIG. 4 TYPE OF SHEARING PUNCH USED

necessary, since the ratio of pressure piston to indicator piston is 2000 to 1.

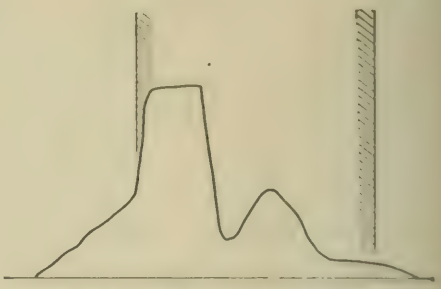
6 Fig. 3 illustrates an oil tank, pump, gage and pipe connections to enable the operator to insure a supply of oil under slight pressure before each operation and to provide for a slight loss through leakage by the indicator piston. The general arrangement of these fixtures is more clearly shown in Fig. 1, which illustrates the mechanism applied to the Olsen testing machine.

7 The motion mechanism for indicating the travel of the punch in the plate is attached to the crosshead of the testing machine or punching machine. It consists of an arc T and a wheel R (Fig. 3), operated by a metallic band connection between the latter and the finger V, which is set in motion by contact with the plate to be punched. The indicator motion ratio is 5.5 to 1.

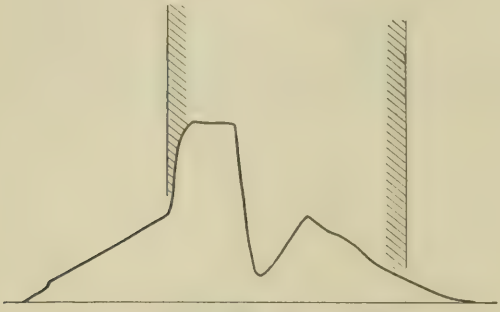
8 Special punches and dies were made for these tests, both flat and shearing punches being used. The latter were of the type illustrated by Fig. 4. No lubricant was used in these tests.



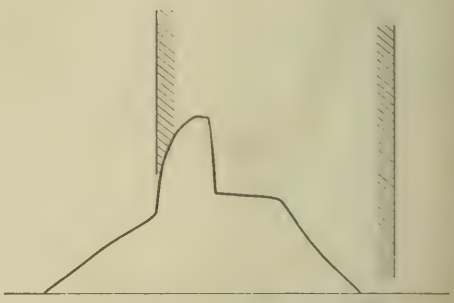
No.1 Die 0.529 Plate 0.257
Punch *F* 400-lb. Spring



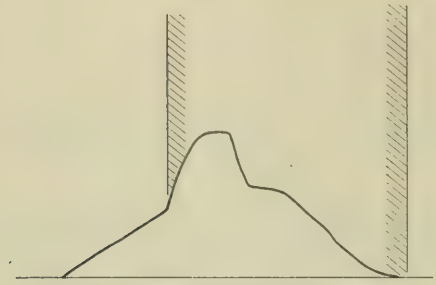
No.3 Die 0.529 Plate 0.257
Punch *F* 400-lb. Spring



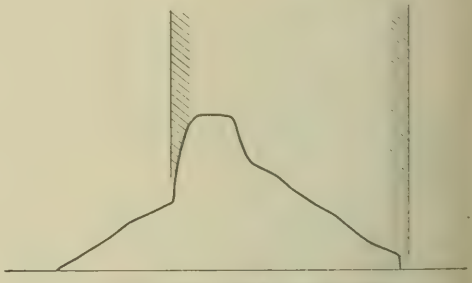
No.9 Die 0.654 Plate 0.257
Punch *F* 500-lb. Spring



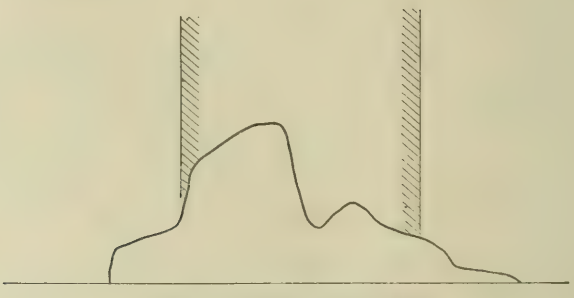
No.38 Die 0.654 Plate 0.257
Punch *F* 500-lb. Spring



No.39 Die 0.654 Plate 0.257
Punch *F* 500-lb. Spring



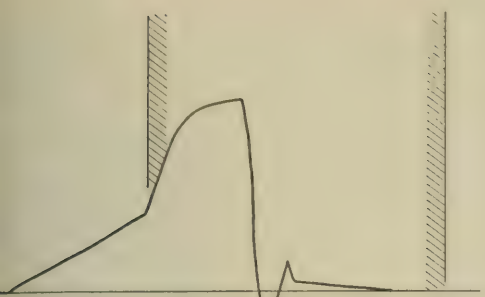
No.40 Die 0.654 Plate 0.257
Punch *F* 500-lb. Spring



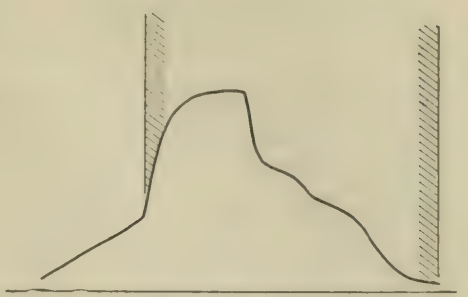
No.7 Die 0.654 Plate 0.257
Punch *S* 500-lb. Spring

FIG. 5 SPECIMEN DIAGRAMS FROM RECORDER

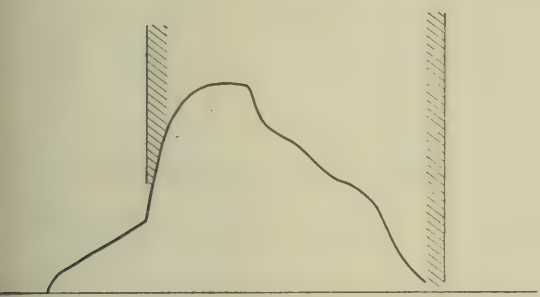
Reduced 30 per cent



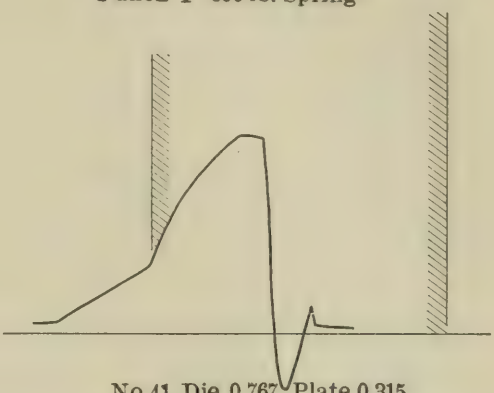
No.33 Die 0.767 Plate 0.315
Punch *F* 500-lb. Spring



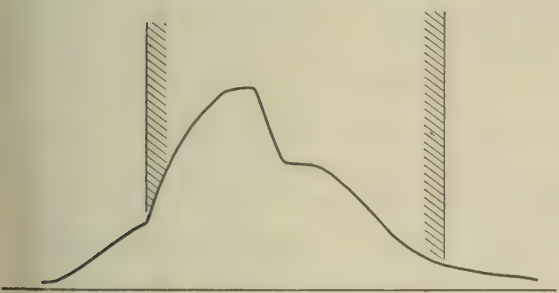
No.34 Die 0.767 Plate 0.315
Punch *F* 500-lb. Spring



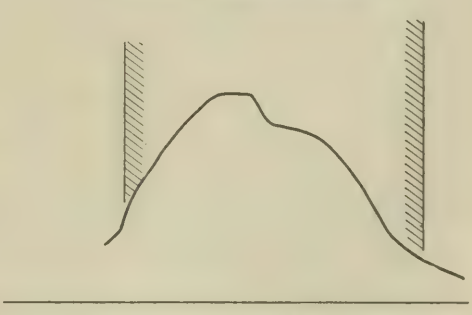
No.36 Die 0.767 Plate 0.315
Punch *F* 500-lb. Spring



No.41 Die 0.767 Plate 0.315
Punch *S* 500-lb. Spring



No.42 Die 0.767 Plate 0.315
Punch *S* 500-lb. Spring



No.43 Die 0.767 Plate 0.315
Punch *S* 500-lb. Spring

No.19 Die 0.767 Plate 0.500
Punch *F* 800-lb. Spring

FIG. 6 SPECIMEN DIAGRAMS FROM RECORDER

Reduced 30 per cent

9 Before taking a card, a datum line of zero pressure was drawn, after which the oil pump was used to obtain a slight pressure in the compression chamber before operating the punch. The series of diagrams shown in Figs. 5 and 6 were reproduced from indicator cards, verticals being drawn in correct relation to the diagrams to indicate the thickness of the plate at the enlarged scale produced by the motion mechanism. Flat punches are designated by *F* and shearing punches by *S*. Cards numbered less than 20 were taken on the Bisbee-Endicott punching machine operated under normal conditions. Cards numbered more than 20 were taken on the Olsen testing machine. When punching on the testing machine the stresses were frequently checked by reading the pressures from the scales of the machine.

10 No. 3 is a characteristic card taken while punching a $\frac{1}{4}$-in. mild steel plate with a $\frac{1}{2}$-in. flat punch on the Bisbee-Endicott machine. The inclined line at the left indicates the pressure required to force the point of the punch into the plate, which in this case is about 1000 lb. The maximum stress is obtained at about 6 per cent of the thickness of the plate and is 23,000 lb. This would make the shearing stress about 57,000 lb. per sq. in. The continuance of the pressure to the right of the plate indicates the force necessary to push the bur through the die. Previous to taking card No. 3 the punching machine was operated by hand to test the adjustment of the apparatus and card No. 1 was taken. In this case the pressure under the diaphragm was raised to about 50 lb. before starting to punch, which will account for the abrupt termination of both ends of the card. The maximum stress is the same as that indicated in No. 3, but occurs at about 15 per cent of the thickness of the plate, due to the increase in the time of the operation.

11 The effect on the maximum stress caused by an increase in the time for the passage of the punch through the plate is shown by cards Nos. 9, 38, 39, and 40, in which the range is from about $\frac{1}{4}$ sec. to $8\frac{1}{4}$ min. There is no apparent cause for the rise in the maximum stress of No. 40 and it is possible that the error is in No. 39.

TABLE 1 EFFECT OF TIME ON MAXIMUM STRESS

No.	Time, Sec.	Maximum Stress	Percentage of thickness of plate at which maximum stress occurs
9	¼	31,000	10
38	80	30,000	18
39	310	25,500	18
40	495	26,000	13

12 Card number 7 shows the effect of using a shearing punch under conditions identical with No. 9.

13 The effect on the stress produced by increasing the clearance between the die and punch is shown by the accompanying table and cards Nos. 33 to 43. The tests were made on the Olsen machine, using a die of diameter 0.767 in. and punching a plate 0.315 in. thick.

TABLE 2 EFFECT OF CLEARANCE ON MAXIMUM STRESS

No.	Type of Punch	Diameter of Punch, In.	Clearance,* In.	Maximum Stress
33	*F*	0.702	0.065	32,000
34	*F*	0.738	0.029	33,000
36	*F*	0.750	0.017	34,500
41	*S*	0.702	0.065	32,500
42	*S*	0.738	0.029	34,000
43	*S*	0.750	0.017	34,000

*Difference in diameters.

14 No. 19 is a card taken while punching a $\frac{1}{2}$-in. plate of mild steel with a $\frac{3}{4}$-in. flat punch on the Bisbee-Endicott machine. This illustrates the effect of a small clearance on the force required to remove the bur, as the difference in diameters of punch and die was but 0.017 in.

ADDENDUM

15 Since the publication of this paper, the pressure-recording device has been quite thoroughly tested by Messrs. G. E. Couillard and W. M. Edmonstone, students of Tufts College, for determining the maximum stresses produced in steel plate, its shearing value, the effect on this value by increasing the clearance between punch and die, and the relative efficiency of various types of punches.

16 At first there arose the question of the error due to the irregularity in the pressure line caused by the viscosity of the oil, and a series of calibrations was made to correct the errors in previous experiments with this apparatus. This use of an indicator has not been an approved one, and our experiments showed too great an irregularity in the pencil movement because of the slow motion of the punch in the testing machine. It was then discovered that a continued tapping of the indicator cylinder during the process of punching eliminated this error most completely, leaving us a constant for the variation in the spring during the upward movement and another con-

stant for the downward movement. As these calibrations were made several times during the punching tests, any slight variation due to the differences between old and new oil was noted. There were such differences, although but slight. With reasonable care in calibration, which in a series of tests is very quickly made, I believe that the error due to the indicator should not be greater than 2 per cent, and this is less than the observed reading on the beam of the testing machine.

17 Another check was applied by connecting a 2000-lb. gage with the oil space. This served for calibration as well as for the reading of maximum pressures.

18 A 600-lb. spring was used for all tests having a maximum pressure of 1220 lb. to the square inch and equivalent to a pressure of 61,000 lb. on the punch.

19 The series of tests alluded to were made on mild steel plate of 60,000 lb. tensile strength, having an elastic limit of about 37,000 lb. Plates of $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$, $\frac{1}{2}$, and $\frac{5}{8}$ in. were used and punches of $\frac{1}{2}$, $\frac{5}{8}$, and $\frac{3}{4}$ in. nominal diameter, variations in these sizes being made to obtain clearances between punch and die diameter of 0.015, 0.030, 0.060.

20 Observations on the shearing stress per square inch, using the minimum clearance between die and punch, and figuring the shearing area as the circumference of the die multiplied by thickness of plate, showed a remarkable difference between the $\frac{1}{4}$-in. plate and all thicker plates, the former having a shearing value of 59,800 while the $\frac{5}{16}$ to $\frac{5}{8}$ plate averaged but 45,800. All curves drawn of the stresses showed great regularity from $\frac{5}{16}$ in. to $\frac{5}{8}$ in. thickness, but the ratio between $\frac{1}{4}$ and $\frac{5}{16}$ being in all cases much greater.

21 The reduction in the maximum pressure due to increasing the clearance between punch and die proved very interesting. Assuming the $\frac{1}{2}$-in. flat punch with minimum clearance, 0.015 as the base, the punching pressure was reduced $7\frac{1}{2}$ per cent by doubling this clearance and 15 per cent by quadrupling this clearance. This percentage was much less with larger punches.

22 Three classes of shearing punches were used, the spiral, an outside bevel, and an inside bevel, and comparative tests were made between these and a flat punch in the punching of nearly 500 holes. On the $\frac{1}{4}$-in. plate the decrease in pressure by the use of a shearing punch varied from 6 per cent with minimum clearance to less than 2 per cent with maximum clearance. On plates thicker than $\frac{1}{4}$-in. there was an increase in the shearing pressure which in some cases

was as great as 34 per cent. Besides this, the difficulties of stripping the plate were greatly increased.

23 Although the device in question does not afford us an opportunity for deriving stripping values, we have made some observations on this subject, and although very incomplete, it is of interest to note that the force required to strip a $\frac{1}{2}$-in. plate from a $\frac{1}{2}$-in. punch varied from 800 lb. to 5 tons, the latter value being with a shearing punch and maximum clearance.

DISCUSSION

R. C. CARPENTER said that although the paper referred to the hydraulic indicator, it seemed to be an ordinary steam indicator of a well-known type, and the author did not state what means were taken to prevent leakage in the indicator, and the consequent reduction of pressure. The drawings indicate that the volume of the fluid which acts on the indicator is very limited, and any leakage would materially affect the results. It is a well-known fact that indicators used under such conditions are very likely to leak or to stick, and either condition would have quite a serious effect.

JULIAN KENNEDY believed that if the die on which the punching was done were mounted on a steel bar of suitable length and fitted with a multiplying apparatus for measuring the deflection of the spring, making a total travel of about $\frac{1}{8}$ in. under the heaviest pressure, it would be possible to obtain more accurate results than with a hydraulic piston. There is no other means of measuring strains so accurate as a tempered steel spring with exact means for measuring its deflection.

THE AUTHOR. As regards the leakage to which Professor Carpenter refers, it has already been stated that the amount is so small that pressure has been maintained for a period of eight minutes, as shown by card No. 40, Fig. 5.

An indicator spring operating under such conditions will, of course, record a lower pressure than when operated rapidly, and must be calibrated for this speed and the fluid used. Irregularities which at first were caused by the sticking of the piston were almost entirely eliminated by the continued tapping of the indicator cylinder, thus enabling us to obtain a very great degree of precision.

A Crosby indicator was used, of the type catalogued as a hydraulic

indicator. We have since checked these results by using a Crosby hydraulic press recording gage in which all leakage was eliminated, and the results from the latter did not differ from the former.

In using the pressure indicator on a punching machine, for which it was designed, the rapidity of the operation suggested the use of an indicator rather than a recording gage.

No. 1320

COMMERCIAL APPLICATION OF THE TURBINE TURBO-COMPRESSOR

By Richard H. Rice, West Lynn, Mass.

Member of the Society

The General Electric Company recently put in operation at the Oxford Furnace, N. J., plant of the Empire Iron & Steel Company, a turbine-driven air compressor (Fig. 1) for blowing the blast furnace, which is the first installation of this type of apparatus to be made in this country.

2 The unit consists of a six-stage compressor operating at a normal speed of 1650 r.p.m. and driven by a direct-connected four-stage Curtis steam turbine. The design is such that this normal speed produces a blast pressure of 15 lb. per sq. in. The unit, however is designed to regulate the volume of air delivered per minute so as to keep the rate of discharge constant at any value determined by the furnace superintendent within its capacity. The manner in which this is accomplished will be fully described in the sequel, but it may be said here that the regulation is by means of speed variations, so that the machine is a constant-volume, variable-speed apparatus, and not a constant-speed as in other classes of blowing units.

3 The compressor has six stages arranged in series, so that the air enters at the end nearest the steam-turbine driver and passes successively from stage to stage until it reaches the other end of the compressor casing, where it enters the discharge pipe. The impeller wheels are so designed that there is no unbalanced end thrust, so that the ordinary means used in the Curtis turbine for locating the rotating elements and preserving proper clearances are sufficient for the entire apparatus.

4 The air is cooled in each stage during compression and also in passing between stages by suitable water chambers in the diaphragms, and this cooling is sufficient to maintain the compression approximately along the adiabatic line.

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

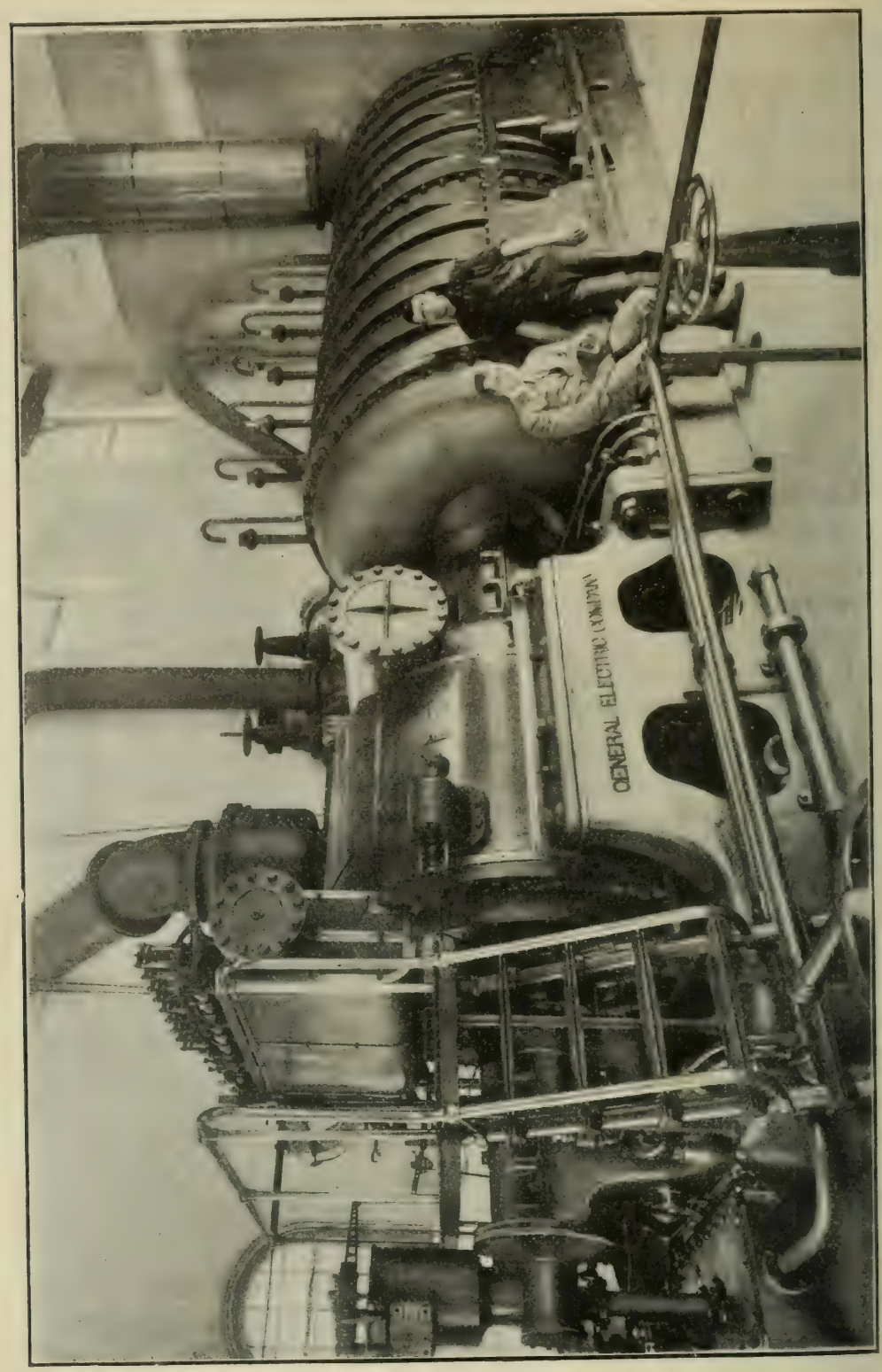


FIG. 1 AIR COMPRESSOR DIRECT-CONNECTED TO CURTIS STEAM TURBINE, 1500 R. P. M. EMPIRE IRON & STEEL CO., OXFORD FURNACE, N. J.

5 No valves or rubbing surfaces are used in the compressor construction and, as in the turbine, the rotating elements revolve freely with ample clearance so that no wear or deterioration can take place; therefore, the efficiency of compression must remain constant.

6 Fortunately, both turbine and compressor attain their best efficiency under similar conditions as regards rotating speed, making the combination a logical and efficient one. Under conditions usually met with in blast-furnace operation involving pressures of blast of 10 to 20 lb. per sq. in., the efficiency remains sensibly the same. A

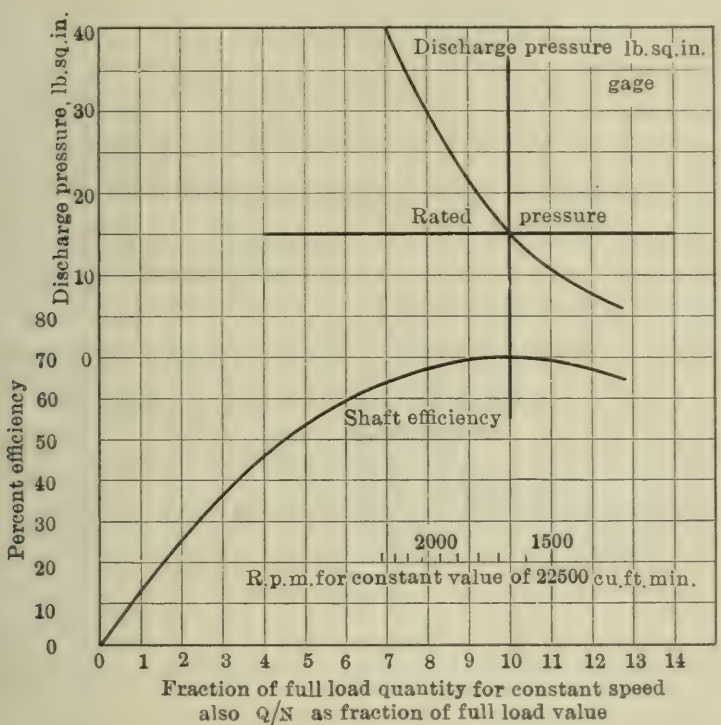


FIG. 2 EFFICIENCY AND PRESSURE CURVE WITH CONSTANT-VOLUME GOVERNOR. COMPRESSOR WITH SIX STAGES

curve of efficiency at varying volumes is shown in Fig. 2 and above this has been drawn a curve of speeds and pressures which, taken in connection with the first named curve, shows the variations of efficiency with pressure, at rated volume.

7 This latter curve shows graphically the variation of pressure with change of speed, which follows the law of squares; that is, doubling the speed gives four times the pressure, etc., from which it will be seen that only moderate changes in speed are necessary to give considerable changes in pressure. It is these changes in speed, increasing or decreasing the blast pressure, which are utilized to main-

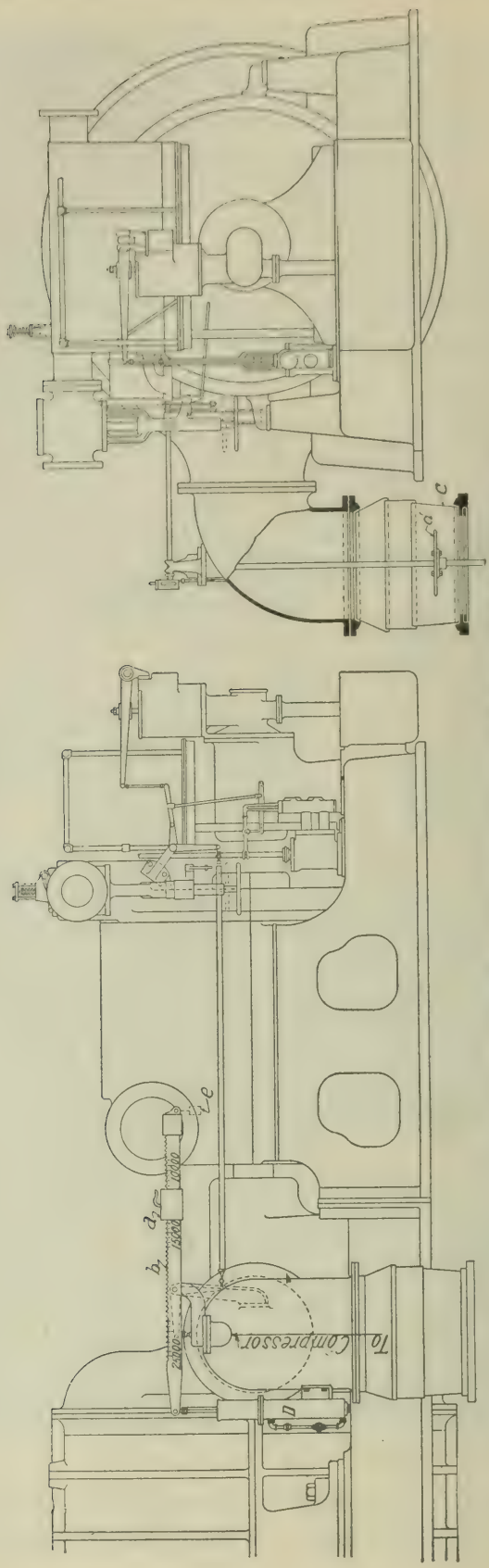


FIG. 3 VIEW SHOWING CONSTANT-VOLUME GOVERNOR

tain a constant rate of flow of air into the furnace, against the varying resistances set up in the tuyeres and furnace by varying furnace conditions; as, for instance, clogging of tuyeres and changes in the size and composition of the charge, temperatures, etc.

8 The means by which these changes of speed are produced in the manner necessary to keep up a constant rate of influx of air per minute is shown in Fig. 3. This shows a steel disc *d* sustained on the inflowing air current within a conical enlargement of the inlet pipe. By means of the sliding weight *a*, the resistance of this float and displacement by the air current is adjusted in accordance with a scale on the scale beam *b* which is graduated accurately in cubic feet per minute to read volumes of free or atmospheric air. By setting this weight at the graduation corresponding to the rate of discharge of air desired, the disk is caused to assume a position in the conical enlargement *c*, which results in supplying steam to the turbine in quantity sufficient to establish the proper speed of the compressor and pressure of blast to cause the required flow of air through the furnace. In case the rate of air flow tends to decrease, the disk *d* sinks to a lower point in the enlargement *c*, since the supporting air current decreases its sustaining power. More steam is by this admitted to the turbine and the speed is increased, resulting in increase of pressure, and this increased pressure reëstablishes the desired flow of air. In case too much air tends to flow into the furnace, the reverse of all these effects takes place. In practice, the operation of this device is most regular and satisfactory.

9 This method of governing, by the indications of a properly calibrated scale beam, gives an entirely new instrument, which, in the hands of a skilled furnace manager, will undoubtedly enable improved results in furnace operation to be obtained, since an accurate knowledge of the amount of air supply is always at hand by this means.

10 Such knowledge cannot be obtained from reciprocating blowing engines, since the expansion of air in clearance spaces causes an error increasing in amount as discharge pressure increases, and because leakage increases with increase of discharge pressure, and the slip is a variable and uncertain amount. On the contrary the air governor is unvarying in its action, and will not change its indications with time, since wear and leakage are absent.

11 It has been intimated before that this is a variable-speed apparatus. In normal blast-furnace operation pressure may vary from 10 to 20 lb. per sq. in. These pressures require speeds in the particular apparatus under description of about 1500 for 10 lb. pres-

sure to 1800 for 20 lb., as appears on the curve, Fig. 2. The blast furnace operator therefore instructs the engineer operating the compressor not to maintain a certain number of revolutions, as is customary with reciprocating engines, but to set the scale beam weight for the required volume in cubic feet per minute.

12 The graduation and calibration of the scale beam in cubic feet per minute is determined during the shop test of the apparatus before shipment by accurate tests with standard orifices and pitot tubes and these graduations are accurate within about two per cent.

13 A simple oil dashpot *D*, Fig. 3, attached to the scale beam, prevents any racing or undue fluctuations of speed.

14 In operating the blowing unit, it is only necessary to manipulate the hand throttle valve in the main steam pipe when it is desired to bring the compressor to rest. At all other times control is effected through the scale beam, with wide-open throttle. At times of checking the furnace or casting, the weight *a*, Fig. 3, is moved to the extreme end of the scale beam at the position indicating the minimum volume for which the scale beam is graduated, and still further decrease of speed and pressure is produced by adding an auxiliary weight at this end of the beam or by depressing it by hand. On removal of the auxiliary weight and replacement of the sliding weight *a*, at the running volume graduation, the compressor speeds up until the volume required is obtained. This manipulation is in practice of the simplest character.

15 The air governor acts upon the pilot valve of the hydraulic valve gear commonly used on the larger sizes of the Curtis turbine, through a system of floating levers, in such wise that when the turbo-compressor nears the maximum speed for which it is designed, in this case 1950 r.p.m., a centrifugal governor of the usual type comes into action and keeps the speed at this maximum as long as the resistance to air flow in furnace or tuyeres remains so high that the volume of air, for which the air governor is set, cannot be forced through at the maximum pressure to which this speed corresponds, in this case 25 lb. per sq. in. During this period the air governor is out of control of speed, but it comes into action immediately when the furnace resistance decreases.

16 In case of breakage or sticking of the governor mechanism which permits the speed to exceed 1950 r.p.m., an emergency governor mechanism, entirely independent of the mechanism previously described, comes into play and closes the main throttle valve, bringing the compressor to rest.

17 In all high-speed apparatus the certainty of the oil supply is an important feature and it is particularly so in this service. In this unit there are three shaft bearings requiring automatic lubrication, and this is furnished by a valveless gear pump, worm-driven from the main shaft, which circulates oil under 15 to 25 lb. pressure. The same pump also supplies this necessary oil to the hydraulic cylinder which actuates the valve gear. The oil is returned from bearings and cylinder to a tank where it is settled and strained before re-use. In order to guard against any stoppage of oil circulation, an alarm is provided which causes a steam whistle to blow in case the oil pressure falls to 5 lb. per sq. in.

18 The oil is cooled in the bearings at the point where the heat is generated, by means of water-cooled coils embedded in the bearing linings.

19 The apparatus described uses, of course, high-pressure steam. Obviously the compressor is adapted equally as well to the use of low-pressure turbines as drivers and so driven affords a ready means of increasing the efficiency of existing plants containing reciprocating blowing engines, by the usual method of exhausting from the reciprocating steam cylinders into the low-pressure turbine. The governing by volume of air discharged is equally applicable here, and all the advantages of this system can therefore be realized.

20 Increased efficiency of the plant to the extent of 20 per cent to 50 per cent may be thus realized with a very moderate addition to the cost.

21 The installation at the Empire Iron & Steel Company, which the photographs accompanying this article represent, was put in operation on the furnace on March 8, 1910, and has been in continuous use ever since. At the time this apparatus was put in operation, it was not expected that the volume of air required by the furnace would be at such a low figure as turned out to be the case, the machine having been designed for a normal volume of 22,500 cu. ft. per min. On putting the machine on the furnace, it was found the volume required was only about 15,000 cu. ft. per min. and the pressure corresponding to this volume under furnace conditions ranged from 10 lb. to 12 lb. Under these conditions, it was found that pulsations were met with in the pressure line, this pressure fluctuating about 2 lb., and in order to overcome this pulsation it was found necessary to throttle the inlet opening. Fig. 4 shows the character and magnitude of these pulsations. Since this time, a convenient butterfly-valve throttling mechanism has been designed and

applied, which is found to eliminate these pulsations without appreciable loss of efficiency.

22 The pulsations in pressure above noted are an inherent characteristic of all centrifugal blowing apparatus of similar construction, and they occur when the apparatus is operated at loads and pressures widely differing from those for which the apparatus is designed; that is, from normal full volume and pressure. At any given volume they occur at a certain critical pressure and at all higher pressures, but do not occur at lower pressures than the critical. As volume is increased, critical pressure increases also. The critical pressure is slightly affected by the density and the humidity of the air.

23 Fig. 5 gives the characteristic critical pressure-volume curve of this compressor.

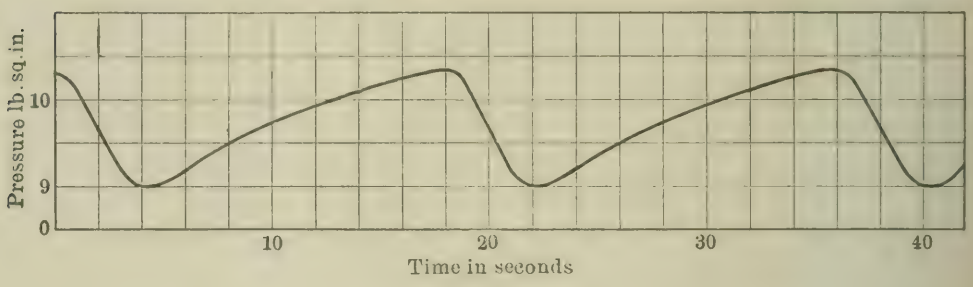


FIG. 4 PRESSURE CURVE DURING PULSATIONS

24 The rate and extent of the pulsations are affected by the capacity of the discharge piping, stoves, etc., into which the air flows. The larger the capacity the longer the period or wave length and the greater the wave magnitude, and vice-versa. The diagram in Fig. 4 shows the pressure waves from the machine installed at Oxford Furnace. When tested in the shop with very short piping of small capacity, the wave length was only a second or so, and of very small height.

25 The pulsations occur only at such loads that the characteristic pressure curve of the apparatus is rising with increase of volume or remains horizontal, and the effect of the throttling is to superpose a drooping pressure curve, falling with increasing volume, which alters the shape of the resultant pressure curve and makes it droop also. As the throttling required to remove such pulsations entirely is only a few inches of water, it has no appreciable effect on the efficiency of the compression.

26 Fig. 6 is the curve of pressure and volumes for this compressor at *constant speed.*

27 The blast pressure at Oxford Furnace varied from 10 lb. to 14 lb. during the day with volume constant at 16,000 cu. ft. per min. The speed varied from 1500 to 1600 r.p.m. The average steam pressure was 135 lb.

28 The figures in Table 1 are taken from a typical station log, showing the variation of pressure and volume during the 24-hr. period of operation.

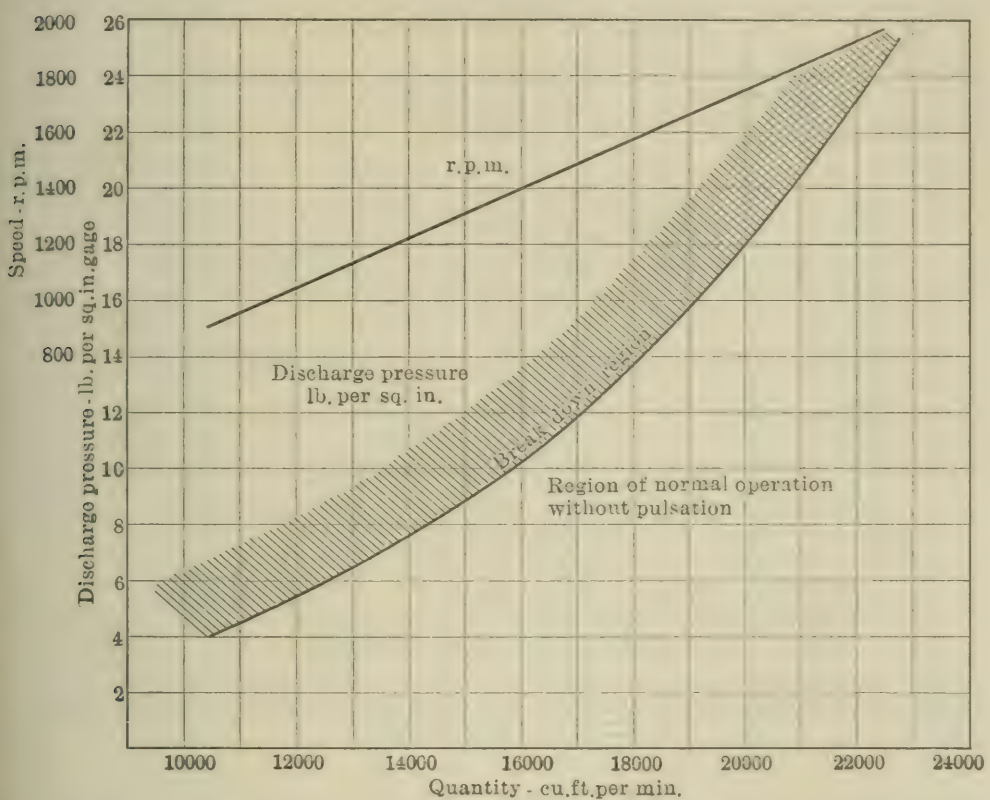


FIG. 5 CURVE OF BREAKDOWN POINTS FROM FACTORY TESTS. PRESSURE AND R.P.M. PLOTTED AGAINST CU. FT. PER MIN.

29 The apparatus used for blowing the furnace before putting this machine into operation consisted of two vertical reciprocating blowing engines built by the I. P. Morris Company, each of the following dimensions: Steam cylinder diameter, 54 in.; blowing cylinder diameter, 72 in.; stroke, 72 in. Blowing cylinder displacement, 339 cu. ft. per revolution each. Maximum speed rating, 30 r.p.m. each, giving 20,300 cu. ft. per min. total displacement. Actual maximum

speed, 23 r.p.m. each, giving 15,000 cu. ft. per min. total displacement. The average blast pressure was 8 lb.

30 Judging from the revolutions of this engine, it was thought that the volume used was about 14,500 cu.ft. On putting the centrifugal compressor into action, an immediate increase in the amount of iron melted by the furnace was experienced. The output went up from an average of 139 tons per 24 hr. in February 1910, to 176 tons in April 1910, and the iron was found to be of a more uniform character and the operation of the furnace was improved. A gradual increase in the amount of air has since taken place and the corresponding increase in pressure required to force the air through the furnace has

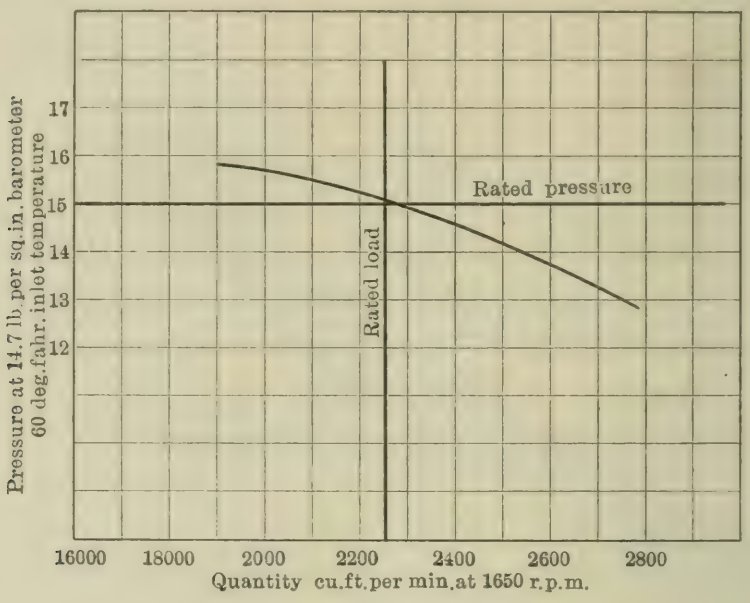


FIG. 6 FACTORY TEST, SHOWING PRESSURE PLOTTED AGAINST QUANTITY OF AIR. 1650 R.P.M.

been necessary, as was to be expected. This increase of air has resulted in an increase in the production of the furnace from 176 tons on starting to the present average of about 190 tons. The machine is now operating with 16,000 cu. ft. of air and the production of ore is 185 tons per 24 hr. average. It is proposed to continue this increase to 200 tons per 24 hr., the limit of the charging apparatus.

31 The dimensions of the furnace are as follows: Diameter at bosh 17 ft. 6 in.; at hearth 11 ft.; at top throat 12 ft.; height from hearth to dumping ring 80 ft.

32 The condensing apparatus is of the barometric type, and the injection water is supplied by a turbo-driven centrifugal pump, placed

in the sub-basement; and at the outset when the machine was first put in operation, difficulty was encountered with the condensing water supply, which made it necessary to operate the machine for a considerable period of time non-condensing. Owing to the unfamiliarity of the fire-room force with the new boilers which had been installed it was even necessary to operate with steam pressures as low as 60 lb. per sq. in. gage for various periods, under which conditions the compressor set operated with entire satisfaction.

TABLE 1 ENGINE-ROOM REPORT, MARCH 17, 1910

EMPIRE IRON & STEEL COMPANY, OXFORD FURNACE, N. J.

Time	Volume, cu.ft.	Blast Pressure, lb.	r.p.m.	Steam Pressure, lb.	Vacuum, in.
a.m.					
1	15750	13	1540	140	24
2	15750	12.5	1490	135	25
3	15750	13.5	1580	135	25
4	15750	12	1510	140	24
5	15750	13.5	1530	155	24
6	15750	13	1550	150	25
7	15750	12.5	1550	150	25
8	15750	12	1490	120	h.p.
9	15750	11.5	1500	130	h.p.
10	15750	13.5	1580	160	26
11	15750	12.5	1520	155	26
12	15750	12.5	1500	150	26
p.m.					
1	15750	13	1530	140	26
2	15750	12	1490	150	26
3	15750	13	1560	130	26
4	15750	13	1560	150	26
5	15750	11	1445	145	26
6	15750	11.5	1450	130	26
7	15750	11	1490	135	26
8	15750	13	1510	140	26
9	15750	12.25	1500	155	25.5
10	15750	11	1383	140	25
11	15750	11.5	1410	150	25.5
12	15750	11.5	1440	145	25

Made 208 tons of iron in 24 hours.

33 Owing to the fact that the condensing apparatus is of the barometric type, the further fact that the machine is operating far below its designed capacity and the difficulties involved in making an accurate boiler test to determine the amount of feed water under present conditions, no tests have been made to determine the actual efficiency of the machine. It is, however, furnishing considerably more air than the old machines, as is evidenced by the greatly increased product of the furnace, and is at the same time operating with fewer

boilers. Also these boilers are more easily worked than when operating with the engine.

34 There is great difficulty in making comparisons of the performance of this type of blowing unit with reciprocating types, either steam or gas driven, owing to the absence of actual test figures, since none have been published which permit of accurate and satisfactory comparison. With the results which have been obtained from all sources as to the actual performance of such machines and from actual experience with this machine and its sister machine installed at the Northern Iron Company, in line with tests which have been made in the factory, it seems that the following conclusions are correct in reference to this apparatus as compared with reciprocating engines for blowing blast furnaces:

a That the output of the furnace is increased on account of the greater steadiness of operation and more uniform conditions obtaining in the furnace.

b That the quality of the product is improved.

c That the steam consumption is equal to, or less than, that of the best compound engines blowing similar furnaces.

d That the engine room space occupied is only a fraction of that needed by reciprocating engines, either steam or gas.

e Considering all factors, including consumption of fuel; cost of operation, including oil and supplies, attendance, etc.; cost of buildings and foundations, interest on the investment; and cost of maintenance of plant; that the centrifugal compressor is a blowing apparatus which can be operated for a lower net cost than any other means of blowing furnaces.

DISCUSSION

E. D. Dreyfus (written). The comparison of the ultimate economies of the reciprocating gas-engine blowing engine, and the high-pressure turbine-driven compressor is most vital. But we must not allow ourselves to be guided by any abstract quantities, and in order to obtain a comprehensive idea of their relative commercial value, we must study a wide range of sizes and conditions. The relation of single small sets of the two types will in all probability be reversed when installations of a great many large units are compared. Isolated data should invariably be treated guardedly, more particularly where an appreciable variation in conditions may exist. There is one thing which occurs to my mind especially and that is

the large gas-engine blower will have but little showing in the extensions of any present mills where it will be practical to install and operate low-pressure turbine-driven compressors in connection with older steam-engine-driven tubs. The investment value of this earlier apparatus will thus be preserved and at the same time its economy when operating in connection with the low-pressure turbine will be greatly bettered. Another point of economic consideration is the coördinating of the low-pressure turbine with the gas plant, whereby the position of the gas engine may be materially advanced, an analysis of which was contributed by the writer to Power, April 11, 1911.

J. E. JOHNSON, JR. The efficacy of the control of the volume of air blown is the question which will arise in the mind of every practical furnace man as soon as he gives this subject consideration. Changes in quantity of air as small as 1 or 2 per cent are frequently made in the quantity of blast delivered to the furnace as one of the means of controlling its operation and this makes it obvious that exact control is a very necessary feature of any blast-furnace blowing apparatus.

I had the pleasure of seeing the installation of the turbo-blower at the Port Henry plant of the Northern Iron Company about a year ago, and the simplicity of the governor is certainly admirable, but is a feature not exactly easy of comprehension to a man without a thorough mechanical training, and a prejudice against these machines is likely to exist in the minds of many furnace men.

It is a well accepted fact that the efficiency of the turbine is far higher at low pressures than it is at high, while on the other hand it is equally well known that for high pressures only, the reciprocating engine has a considerable advantage over the turbine. The result is that the most economical form of steam-driven apparatus is a combined unit in which the steam expands down to about atmospheric pressure in a good reciprocating engine, and is then delivered to a turbine on its way to the condensers. The very remarkable results obtained with this combination by Mr. Stott at the Interborough plant in New York have given, in the opinion of many, a decided check to the development of the gas engine.

In a similar way it has been known for years that for very low pressures the centrifugal blower, of which the turbo-blower is simply a development, is more economical than a piston blower. On the other hand, for higher pressures the piston compressor is undoubtedly to be preferred, particularly in blast furnace work, on account of its operating as a meter as well as a compressor of the highest efficiency.

A condition of piston blowing engine design is that the air cylinder must be large enough for the greatest volume to be handled, and strong enough for the highest pressures attained, with the result of enormous size and very massive and expensive construction in modern blowing machinery.

In spite of their high cost it is notorious among operating men that really reliable blowing engines are exceedingly scarce. The problem of the valves is in itself one of great difficulty. It is almost impossible to get an air inlet valve with sufficient area to allow the cylinder to fill absolutely without heavy loss by suction, and at the same time one quick enough to give correct results at the speeds required of modern engines.

The turbo-blower, on the other hand, suffers from the disability that it must have stages enough to blow the highest pressure ever required, although this high pressure will not, in ordinary operation, be required one per cent of the time, the normal pressure being perhaps only half as great.

If, therefore, we make a turbo-blower to deliver air of a certain nearly constant, moderate pressure, and a blowing cylinder of comparatively small size, strong enough to deliver air to the desired maximum pressure, we shall have bettered the conditions of operation of both types of apparatus, increasing their efficiency and reducing their cost. In other words, the turbo-blower and reciprocating-compressor combined are more efficient than either apparatus alone, exactly as the engine-turbine combination is more economical in the consumption of steam.

For the best and cheapest blowing engine, therefore, we should have a combination of these two units, a turbine-driven turbo-blower, delivering air partly compressed to the compressing end of a steam-driven blowing engine, the steam from the cylinder of this engine driving the turbine before passing to the condenser.

It may be objected that this is a complicated unit, but this is not the case. Most large modern furnaces require three large reciprocating blowing engines of the best type to operate them. These are, in good modern usage, disconnected compounds—a high-pressure steam cylinder on one engine and a low-pressure on the next, etc.

A turbo-blower supplying air to two of these, compressed to 8 lb., would enable them to deliver as much air as three engines drawing air from the atmosphere, and the efficiencies which are guaranteed in the use of exhaust steam by the builders, combined with the guaranteed efficiency of the turbo-compressor, indicate that this can be done at

a decided saving in first cost and with an efficiency certainly no lower and probably higher than that of either unit alone. This does not make any allowance for the possibilities of spray intercooling between the turbo-compressor and the reciprocating compressors, which, in ordinary practice, would be good for about 4 per cent additional saving at a very low cost.

The advantage of maintenance as compared with reciprocating engines, is undoubtedly with the turbo. At the same time the magnitude of the alternations of stress in the reciprocating engines is greatly reduced by delivering to them partly compressed air, and the problem presented by their inlet valve gear, which has given so many bad hours to both furnace men and engine designers, is practically eliminated by delivering to them denser air under pressure.

The governing of such a combined unit is extremely simple. The ordinary governor on the engine unit does all that is necessary and no governor other than one to prevent racing is necessary on the turbine. The steam supply of the engine passes directly to the turbine. As long as all conditions remain constant the speed of the turbine will be unchanged, but if the pressure required by the furnace increases a little, more steam is admitted to the engine by its governor, and this increase in quantity speeds up the turbine to a slight extent and so causes it to deliver air compressed to a slightly higher pressure.

This increase automatically compensates for any small lag due to the greater load on the engine and for the lower volumetric efficiency of the piston blowing engine at higher pressure as compared with low.

If, on the other hand, the speed of the piston engine is deliberately increased to deliver more wind, the greater supply of steam to the turbine will enable it to deliver a proportionately greater quantity of air at virtually the same pressure. Of course, some hand control will be applied to the turbine to enable the receiver pressures to be adjusted to the best conditions, but once this is done other changes will seldom be required.

For these reasons I believe that the steam-driven blowing plant of the future should consist of a turbine-turbo, piston-engine, piston-compressor combination.

This combination will have its greatest field in applications to existing plants where, as is so often the case, the existing blowing engine, while old, is by no means worthy of the scrap-heap, but is unable to deliver quite the pressure or quite the volume demanded by modern conditions. In such a case the introduction of a turbine-

turbo, taking steam from the engine and supplying pre-compressed air to it, will use what is now waste energy in the steam, and apply it in such a way as to increase both the capacity and the permissible pressure of the existing blowing cylinder.

A further advantage which the turbo has as the primary machine is that of an inlet free from pulsations. These are much more serious in inlet than in outlet, for the reason that the volume and velocity of the air are so much greater.

One of the serious problems which had to be overcome in the application of the dry blast was the construction of cooling chambers whose walls would not be thrown down by the pulsations produced by the piston compressor on the incoming air. The smooth inlet of the turbo will eliminate entirely difficulties arising from this source.

The necessity of massiveness both in stationary and in reciprocating parts, and the conditions of operating the air valves, have limited the best sizes of blowing engine so that these are not increasing in size. In fact, those now being built are commonly no larger than some constructed ten or twelve years ago.

The turbo-blower, on the other hand, lends itself admirably to enormous capacities in the same way that the turbine is commonly built for certain classes of service in sizes that have never been approached in reciprocating engines. For this reason it will not only be possible but advisable to install at large plants one turbo to supply several reciprocating engines, all of which, of course, will exhaust to it. This will contribute still more to low first cost, small space requirements and small maintenance charges.

I wish to express in conclusion my conviction that the introduction of the turbo-compressor marks the beginning of a better era for the mechanical equipment of blast furnaces, but that its best and easiest application will be in conjunction with, rather than in supplanting piston blowing engines.

R. N. Ehrhart.[1] The Westinghouse Machine Company have for some years been experimenting with centrifugal air compressors at their East Pittsburgh works, and as a result have developed a type of blower possessing features particularly adapting it to certain classes of work.

As the single-stage blower is the basic element of the multi-stage blower, analyses of the primary considerations should be applied to

[1] The Westinghouse Machine Company, East Pittsburgh, Pa.

the single part or stage which is duplicated in certain instances to make the multi-stage outfit. Any features pertaining to the generic element pertain to the whole, so that studies of the single-stage blower are more enlightening from a practical and educational standpoint. The following facts were brought out in the development of the elements of centrifugal compressors:

Centrifugal compressors having diffusion vanes in the part of the casing receiving the air from the rotating element have as an in-

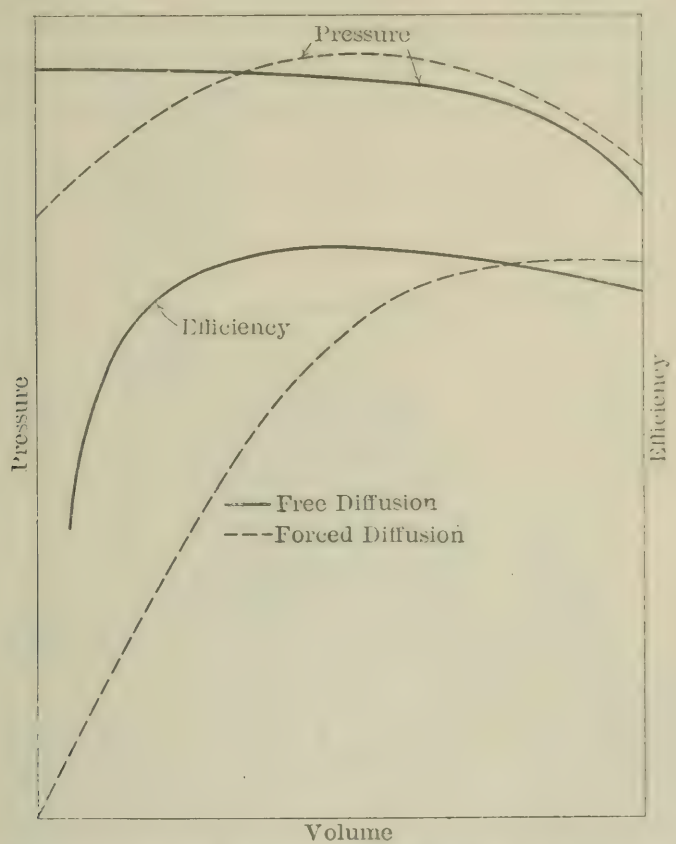


FIG. 7 COMPARISON OF BLOWERS WITH FREE AND FORCED DIFFUSION

herent characteristic the pressure curve shown in broken line in Fig. 7. (In this discussion, the writer calls this the forced-diffusion type of blower.) The efficiency also has the characteristic as shown in Fig. 7; that is, a maximum efficiency of approximately 74 per cent at rated volume and approximately 60 per cent at one-half volumetric rating. These results were obtained by extensive experimenting at East Pittsburgh and are approximately verified by Fig. 2.

For the industrial application of constant-speed blowers, it was early realized by the Westinghouse Machine Company that the

pressure characteristic as shown by the broken line (forced diffusion) in Fig. 7 was unsuitable, as reference to Fig. 8 clearly shows.

The forced-diffusion type of blower has a pressure characteristic with a well defined hump; that is, the pressure rises as the volume delivered increases to a certain point, and then rapidly falls off. In other words, for a given pressure there are two definite volumes that may be delivered. Referring to the lower set of curves, Fig. 8, at the pressure represented by the horizontal line, there may be two volumetric deliveries, *B* and *D*. It is perfectly evident that blowers of the forced-diffusion type will be unsatisfactory for parallel operation, unless equipped with a complicated governing mechanism, since momentary disturbances may cause fluctuating line pressures

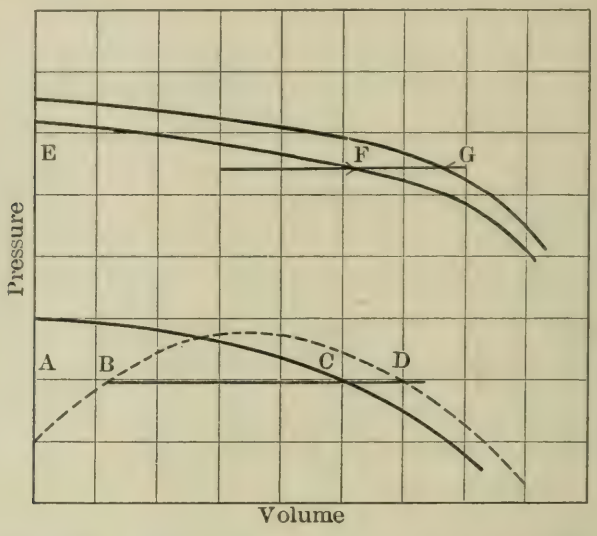


FIG. 8 CHARACTERISTICS REQUIRED FOR PARALLEL OPERATION OF BLOWERS

and if the disturbance has a magnitude measured from the horizontal line to the peak of the pressure characteristic, the volumetric delivery may vary from *B* to *D*. This latter possibility will in turn accentuate any line disturbances, causing still greater variation or fluctuation in volumetric delivery, assuredly at the sacrifice of efficiency.

In the case of parallel operation of alternators, we require a gradual drop in speed from no load to full load for stability. In the parallel operation of blowers, we should have the analogy of a gradual drop in pressure from no delivery to full delivery.

Blowers having this characteristic will work perfectly in parallel. Referring to Fig. 8, the upper pair of lines, shows the sharing of the load. One blower at the pressure represented by the horizontal line

will have a volumetric delivery *EF*, the other one a volumetric delivery *EG*, and fluctuations of pressure over a wide range will still insure relatively equal division of the load.

Positive blowers have the drooping characteristic at constant volume, and it is seen that if this is superimposed on the forced-

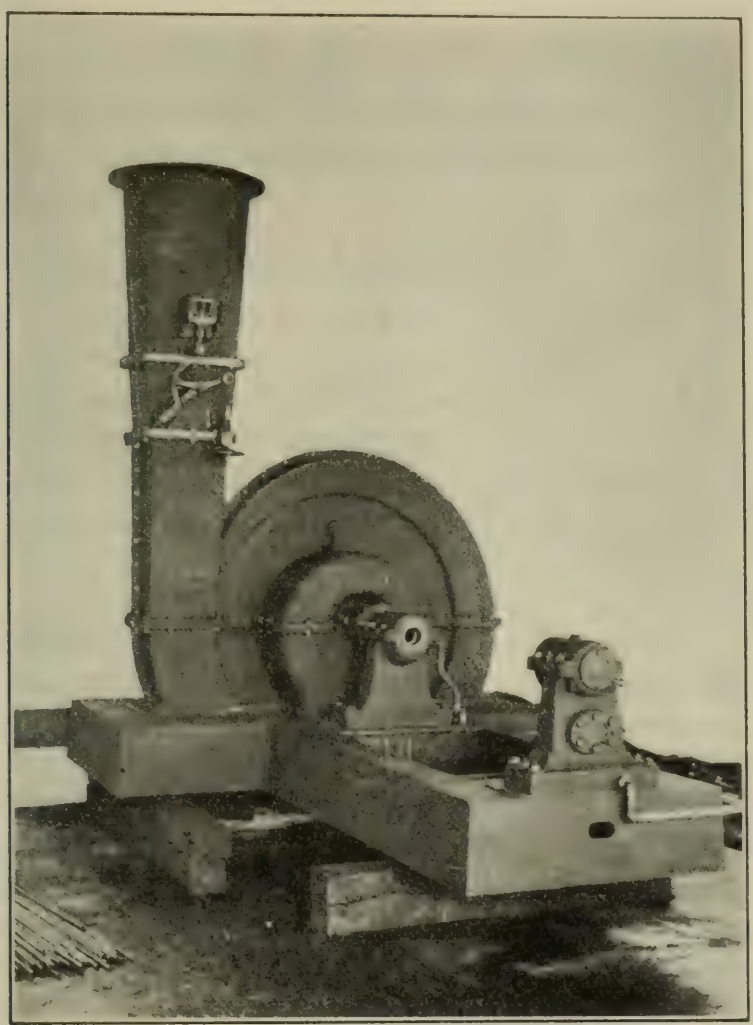


FIG. 9 A 600-H.P. BLOWER OF FREE-DIFFUSION TYPE

diffusion pressure characteristic, successful parallel operation cannot be had.

Of course, by resort to complicated speed-control mechanisms on what might be a constant-speed outfit, the forced-diffusion type of blower can be made to give the drooping characteristic.

Thorough experimenting has shown that by suitably constructing the stationary and rotating parts of the blower, a drooping pressure

characteristic can be obtained; that is, the pressure at no delivery is a maximum, and it gradually drops off to normal at normal rating. This characteristic is shown in full line in Fig. 7. It has also been found that the efficiency at fractional deliveries was wonderfully improved. As a matter of fact, the diffusion vanes in the forced-diffusion type of blower are a positive detriment at fractional loads, since their angle and shape can be made only to suit one discharge rate.

For example, in a blower with radial vanes, the air leaves the rotor at an angle of about 14 deg. with the tangent at normal delivery. At 50 per cent of normal delivery the angle is about 7 deg.; at 10 per cent of normal delivery it is about $1\frac{1}{2}$ deg. It is apparent that if the receiving ends of the diffusion vanes are set for normal delivery, they are $12\frac{1}{2}$ deg. from the correct position for 10 per cent of normal delivery, or 7 deg. from correct position for 50 per cent normal delivery.

Experiments at East Pittsburgh show that free spiral diffusion can be obtained with substantially the same conversion efficiency at 25 per cent normal delivery as at full normal delivery. The discharge areas and contour of vanes and receiving sides of the spiral whirl chamber must coincide exactly with the dimensions as fixed by experiments and calculations. Relatively small variations from proper shapes cause loss of velocity and conversion efficiency.

Fig. 7 shows comparative tests of the two types of blowers made at the Westinghouse works, the splendid pressure characteristic and superior efficiency of the free-diffusion type at fractional loads being clearly illustrated.

The free-diffusion type of blower does not have the pulsation characteristic that the forced-diffusion type has. This is due to the total absence of diffusion vanes, so that free spiralic diffusion can adapt itself to any volumetric delivery. The pulsations as described by Mr. Rice in Pars. 22 to 25 (Fig. 4), would seem to be of such magnitude (approximately 9 to 10.5 lb.) as to make parallel operations difficult when they are in evidence. The method used at Oxford to get over the pulsating effect, that is, throttling, may be successful when applied to high-pressure blowers, but when applied to blowers of low pressure the loss of a few inches of water pressure by throttling could not be considered as it would represent too great a proportionate loss in efficiency.

A 600-h.p. blower of the free-diffusion type has just been put into operation at Great Falls, Mont. The pressure characteristic corresponding to that given in Fig. 7 is attained with a constant speed,

and no governing devices to change the motor speed are necessary. The outfit is driven by a 750-h.p. synchronous, 60-cycle motor. Fig. 9 shows the blower by itself.

C. G. DE LAVAL. The development of the turbo-compressor in this country is of recent origin and without question the combination of the steam turbine and turbo-compressor produces an ideal unit. It appears, however, that its field will rather be the compressing of large than of small volumes. Not long ago it was considered unsuitable for anything except exceedingly low pressure, such as ordinary blowing work. Today we find turbo-compressors in use abroad for 150 lb. equal in thermal and mechanical efficiency to reciprocating compressors.

It is well to consider the distinction between a blower and a compressor, and it appears to the writer that the machine described is not a compressor, but a turbo-blower. Both may be of the same design, but this is not essential. The machine described is a multistage blower or fan, and apparently has no rubbing surfaces. The rotating surfaces revolve freely with ample clearances.

In turbo-compressors manufactured by Pokorny and Wittekind, Brown-Boveri & Company, Escher Wyss, Sulzer Brothers, Rateau and others, the machines are made up in multiple cylinders according to volume and pressures. The general principle consists of high-speed impellers incased in a cylindrical chamber, properly cooled radially and axially. These impellers draw in and discharge the air the same as a centrifugal pump impeller, turning the velocity into potential energy in the form of air pressure.

It does not appear necessary to use six stages arranged in series for so low a pressure as 10 to 20 lb., and if the compressor end were divided into separate cylinders or casings, each cylinder or casing containing one or more impellers, and the diameter of the impeller and its casing were reduced in the second or third, the number of stages required would be less than that used by the author, provided also that proper cooling was effected radially as well as axially.

A turbo-blowing engine built on the Rateau type for 12,300 cu. ft. of free air per minute down to 6000 cu. ft. for a pressure of from 8 to 13 lb., speed 3400 to 3900, requires three stages only as against six stages of the one described in the paper. This Rateau multi-blower has single entrance impellers, leading the discharge from one wheel into the eye of the other through special channel castings. For pressures of 80 to 150 lb. it is not necessary, if it is properly con-

structed, to use more than 10 to 15 stages, and for lower blowing pressures, 10 to 20 lb., 3 stages, and the blower will still maintain an isothermal efficiency of 70 per cent. Such compressors will have a weight about one-quarter of that of a reciprocating outfit.

It has been shown through their use in Europe that turbo-compressors with capacities of 6000 cu. ft. per min. and above have from 70 to 80 per cent efficiency, and those of from 3000 to 6000 cu. ft. per min., 65 to 70 per cent efficiency, and that these efficiencies decrease below 50 per cent at $\frac{1}{2}$ to $\frac{1}{4}$ load. The application of turbo-compressors should be for volumes from 3000 cu. ft. per min. and up.

It is usual to fit compressors with an automatic outlet valve in order to prevent any detrimental pulsation of air at partial capacities, so that the capacity can be kept above the minimum. To prevent an increase of the capacity over the maximum, an automatic valve is installed in the suction.

The use of a turbo-blower in connection with existing reciprocating compressors has been carried out lately in England. The existing compressor was coupled up to a steam turbine turbo-blower, whereby the low-pressure steam cylinder of the reciprocating air compressor exhausted into the steam turbine, and the turbo-blower took the air at atmospheric pressure, compressing it and discharging it into the reciprocating compressor cylinders where it is compressed to 60 lb. pressure.

This combination of turbo and piston-compressor doubled the capacity and gave a net gain of 17 per cent over that which would have been secured had an additional reciprocating compressor been put in. In addition to this it saved the extra cost of a new building on account of the small space it occupied and increased the over-all efficiency, as it utilized the exhaust steam.

It may not be out of order to add the following as characteristics of turbo-blowers and compressors:

a Steady non-pulsating currents when fitted with proper control apparatus.

b Smaller weight and space required, reduction of foundations and handling.

c Absence of reciprocating parts, valves and packings.

d Smaller attention and maintenance.

e Reduction of oil consumption and supply.

f Continuous and steady delivery of air, flexibility of operation, automatic arrangement.

g Utilization of exhaust steam from any reciprocating engine to run the low-pressure turbine operating the blower or compressor.

h Efficiency of the machine, comparing favorably with the best reciprocating compressors.

i Adaptability for cupolas, brass furnaces, converter, operating pneumatic tools, etc.

j Adaptability to operation and furnishing of illuminating gas to mains.

JULIAN KENNEDY[1] said he did not know much either about gas engines or turbines, but he had been doing a little figuring. On a basis of $116 per kw. for gas-engine installations and $60 for steam-turbine installations, he assumed that 20 per cent was a fair amortization charge for up-keep and capitalization for the gas machinery, and 15 per cent for the steam machinery: the investment cost, therefore, for gas machinery would be $23.30 per year, and for the steam machinery $9 per year.

Assuming 7200 hours per year and 2 lb. of coal per kw-hr. at $1.80 per net ton, the cost of coal per kw-hr. would be $0.18, or less than the additional charge of up-keep and capitalization for the gas-engine installation. In other words, according to his figures, if furnace gas were obtained for nothing, and if coal was $1.80 a ton, the result at the end of the year would come out about even.

In his figures he put in furnace gas free, so that if the engine could be brought to such a point of economy as to run without gas, his calculation would hold good provided it cost 5 per cent more to keep up the gas engines than the steam turbines; but he thought 15 per cent would be getting nearer to it.

JOSEPH MORGAN called attention to the fact that the value of the surplus gas and the power generated from it depend entirely on how much can be sold. If there is no market for the power, the gas is worth nothing, while the value of a coal pile remains until used.

C. J. BACON and H. J. FREYN. An analysis (Table 2) of the financial considerations involved in a comparison of a turbo-blower equipment with a gas blowing-engine equipment for blowing a blast-furnace plant consisting of four 500-ton furnaces, leads to results indicating that the total expense for fuel, operation and fixed charges

[1]Consulting Engineer, Pittsburgh, Pa.

is less for a turbo-blower installation than for a slow-speed gas blowing-engine installation in localities where the price of coal establishes a value lower than approximately $0.08 per 1,000,000 B.t.u. in the blast-furnace gas; and is less for turbo-blowers than for high-speed gas blowing engines where the price of fuel is lower than approximately $0.05 per 1,000,000 B.t.u. This conclusion is based on normal conditions prevailing in the average blast-furnace plant, but is influenced considerably by blast pressure, efficiencies, price of labor, water supply for condensers, value set on gas, cost of machinery and construc-

TABLE 2 TOTAL EXPENSE INCLUDING FUEL, OPERATION AND FIXED CHARGES

Cost of Fuel, Cents per 1,000,000 B.t.u.	Turbo-Blowers		Gas Blowing Engines			
			Slow-speed		High-speed	
	Yearly	Per 1,000,000 Cu. Ft.	Yearly	Per 1,000,000 Cu. Ft.	Yearly	Per 1,000,000 Cu. Ft.
			15 lb. Blast Pressure			
4	276700	3.70	340400	4.49	292460	3.85
6	324600	4.34	360000	4.75	312060	4.11
8	372100	4.97	379500	5.00	331560	4.37
10	420100	5.61	399100	5.26	351160	4.63
12	468100	6.25	418700	5.52	370760	4.88
14	516100	6.90	438200	5.77	390260	5.14
			20 lb. Blast Pressure			
4	300100	4.06	350100	4.61	302160	3.98
6	359100	4.80	374500	4.94	326560	4.30
8	419100	5.60	398900	5.26	350960	4.62
10	478100	6.39	423300	5.58	375360	4.94
12	537100	7.17	447700	5.90	399760	5.27
14	597100	7.98	472100	6.22	424160	5.59

tion material, rate of interest, taxes, insurance, estimates of life of apparatus, method of computing depreciation, etc.

For blowing four 500-ton furnaces there would be required eight gas blowing engines, including two spare engines, each with sufficient capacity for supplying two-thirds of the air required per furnace at a normal pressure between 15 and 20 lb. gage and occasionally as high as 25 lb. The usual displacement for a 500-ton furnace is 40,000 cu. ft. per min., although the amount of air actually discharged is somewhat below this figure and is considered as 89 per cent or 35,600 cu. ft. on the basis of actual measurements. Slow-speed engines are supposed

to have a displacement of approximately 500 to 600 cu. ft., and high-speed engines approximately 360 cu. ft. In the case of the turbo-blower installation, six units would be required, including two spare units, each of 35,600 cu. ft. per min. actual discharge capacity at the same pressures. A convenient basis for comparing the costs of operation of different installations is per 1,000,000 cu. ft., in which connection each of the plants under consideration has a total yearly capacity of 74,845 million cu. ft. at 35,600 cu. ft. per min. per furnace.

In preparing the above figures it is assumed that the entire plant of four furnaces would operate continuously at the rated capacity of 500 tons per day. In case of reduced output or of partial shutdown due to dull business, relining furnaces, etc., the reduction in the item of fuel expense would be about twice as great with turbo-blowers as with the gas engines, while the item of fixed charges would continue at the same rate, resulting in a greater net decrease in the total expense for the turbo-blowers than for the gas engines.

Moreover, as certain approximately equal percentage allowances are made in the various efficiencies leading to a comparison of the heat consumption to take account of the actual operating conditions over long periods, any change in the fuel cost due to over-estimating such allowances would also result in a greater net decrease of total expense of the turbo-blowers than the gas engines.

Fuel Expense. Blast-furnace gas, after being subjected to preliminary cleaning, is burned under boilers for furnishing steam power for the turbo-blowers which are direct connected to high-pressure steam turbines. For the gas blowing engines the gas is given both primary and secondary cleaning. In each case the same value is set on the B.t.u. contained in the raw or uncleaned gas according to the local price of coal. This is an equitable basis since a furnace plant of this size would be in close proximity to steel mills in which the power produced from surplus blast-furnace gas is entirely used, thereby displacing coal which otherwise would be burned. The gas is therefore not a waste product.

Since the B.t.u. per lb. of coal, as well as the price, vary greatly in different localities, a unit of cost of fuel per 1,000,000 B.t.u. is used to facilitate comparisons. In the Pittsburgh district the cost is between $0.035 and $0.05 per 1,000,000 B.t.u.; in Ohio, $0.05 to $0.065; in Illinois and Indiana about $0.08.

The heat consumption for blowing the furnace plant under consideration involves a comparison of the various efficiencies affecting the

conversion of the heat in the gas into power as represented by the actual delivery of compressed air. In assigning values, the results of actual performance under operating conditions are used in all cases where applicable data are available (Table 3).

TABLE 3 DATA AND METHOD USED IN OBTAINING THE FUEL EXPENSE

Type of Equipment	Turbo-Blower		Gas Blowing Engine	
Blast pressure, lb. per sq. in	15	20	15	20
M.e.p. of adiabatic compression, lb. per sq. in	11.4	14.2	11.4	14.2
M.e.p. in air tubs of blowing engine, lb. per sq. in	.....	.....	12.1	15.1
Efficiency of compression, per cent	.....	.....	94	94
Displacement of tubs per furnace, cu. ft. per min	.....	.....	40000	40000
Volumetric efficiency, per cent	.....	.....	89	89
Air delivered to blast mains, cu. ft. per min	35600	35600	35600	35600
Theoretical air h.p. per furnace	1775	2210	1775	2210
Air h.p. of blowing engine per furnace	.....	.....	2115	2670
Mechanical efficiency of tub, per cent	.....	.....	92	92
Shaft efficiency, per cent	70	70	77	77
B.h.p. per furnace	2536	3156	2304	2869
Steam consumption, lb. per h.p-hr	12.5	12.5	.....	.....
Thermal efficiency at shaft, per cent	9.46	9.46	21	21
Thermal efficiency at theoretical air h.p., per cent	6.62	6.62	16.2	16.2
Heat consumed per furnace, 1,000,000 B.t.u. per hr	68.2	84.8	27.9	34.8
Yearly cost of fuel per 1,000,000 B.t.u for four furnaces				
At $0.04	$ 95600	$119000	$ 39100	$ 48800
At 0.06	143500	178000	58700	73200
At 0.08	191000	238000	78200	97600
At 0.10	293000	297000	97800	122000
At 0.12	287000	356000	117400	146400
At 0.14	335000	416000	136900	170800
Cost of fuel per 1,000,000 B.t.u. per 1,000,000 cu. ft. air				
At $0.04	$1.28	$1.59	$0.52	$0.65
At 0.06	1.92	2.38	0.78	0.95
At 0.08	2.55	3.18	1.05	1.30
At 0.10	3.91	3.97	1.31	1.63
At 0.12	3.83	4.75	1.57	1.96
At 0.14	4.47	5.56	1.83	2.28

Notes on the Turbo-Blower. The shaft efficiency, or the relation of the power in the discharged air to the shaft power for the particular turbo-blower under discussion is shown to be 70 per cent. Recent developments in larger sizes, it is claimed however, have resulted in as high as 72.5 per cent, but as there is no assurance as yet that the efficiency can be continuously maintained at that point under actual operating conditions, 70 per cent is taken as a fair figure for this comparison.

The steam consumption is taken at 12.5 lb. per b.h.p-hr. At 135 lb. gage pressure, 125 deg. superheat and 28.5 in. vacuum and an efficiency referred to the Rankine cycle of 60 per cent, which may be

expected from a turbine of this size, the steam consumption would be 11.9 lb., which is increased to 12.5 due to the effect of variation of operating conditions.

A boiler efficiency of 62.5 per cent is the actual performance shown by numerous tests of gas-fired boilers with superheaters operating under the normal conditions prevailing at a blast-furnace plant. It is estimated that 8 per cent of the total steam generated goes to auxiliaries of boilers and turbo-blowers and to miscellaneous losses. The boilers for the turbo-blowers are considered as operating at 140 lb. gage pressure, 150 deg. superheat and 70 deg. hot-well temperature. Under these conditions, the thermal efficiency of the turbo-blower installation complete would be 9.46 per cent at b.h.p., or 6.62 per cent at h.p. of adiabatic compression.

Notes on the Gas Blowing Engine. The shaft efficiency, or the relation of the power represented by the actual volume of compressed air to the power required at the shaft, consists of compression, volumetric and mechanical efficiencies.

The compression efficiency is the ratio of the work required adiabatically to the work actually done in the blowing tub. The losses consist principally of those due to inertia and mechanical friction of the discharge valves, to the friction of air through the passages and to late opening of suction valves when such occurs. These sources of loss, taken together, amount to an increase in work of 15 to 18 per cent over theoretical adiabatic compression for certain types of pot valves which require considerable excess pressure for opening. The thin plate automatic valves on Slick tubs show very small losses; in fact, tests made on Snow engines show less actual work than theoretical adiabatic compression. Radiation and water-cooled heads have a varying effect on the power requirement. The results of tests on a number of different kinds of blowing tubs show an average loss of about 6 per cent or an efficiency of compression of 94 per cent for the usual blast pressure of 15 to 20 lb., but varying from 82 to 94.6 per cent within the range of 5 to 30 lb. pressure.

The volumetric efficiency is the ratio of the measured volume of air actually delivered (reduced to atmospheric pressure) to the volume of piston displacement. The losses consist of: (*a*) those shown on indicator cards due to the effect of clearance and friction of air through suction valves, which usually range between 0 and 5 per cent; (*b*) leakage of air past the suction valves, discharge valves and piston; (*c*) the effect of preheating the incoming air during the suction stroke by contact with the surface left hot by the preceding discharge stroke. Tests made on seven blowing engines, part with Southwark tubs and

part with Allis-Chalmers tubs having Reynolds valves, show volumetric efficiencies varying from 79.2 to 89.4 per cent, with blast pressure ranging from 10 to 20 lb. approximately. As some of the engines tested had been in service a number of years and were consequently not of recent design nor in first-class condition, it is not reasonable to consider the results obtained therefrom in establishing a rational figure for tubs of modern design and in fair condition. Volumetric efficiencies of 85 to 98.4 per cent were obtained from Allis-Chalmers tubs with Reynolds pot discharge valves, and as this engine had been in service between two and three years, it may be considered in fair condition. Consequently, in this discussion 89 per cent is taken applying at 15 lb. blast pressure, but varying from 91 to 86 per cent at 5 and 30 lb. blast pressure respectively.

The mechanical efficiency of the air tub, or the ratio of the work shown on the air card to the work required at the shaft, ranges from 86.8 per cent at 5 lb. pressure to 95.8 per cent at 30 lb. pressure. Tests on the Snow engines with Slick tubs show 500 h.p. friction loss between the gas and air cylinders, when blowing 35,400 cu. ft. displacement, or 60 r.p.m. at 15 to 20 lb. blast pressure. The actual horsepower developed in the gas cylinders was 2400, while 1900 was shown in the air cylinders, or 500 h.p. loss. As the speed during the tests was greater than necessary for blowing two-thirds of the requirement of a blast furnace, the friction is taken at 420 h.p., the actual loss shown at the required speed. Taking 420 h.p. as the average total friction loss between gas and air cylinders, the corresponding mechanical efficiency is 79 per cent at 1980 i.h.p. which the Snow tests showed necessary in the gas cylinders for 45 r.p.m., or 26,670 cu.ft. displacement at 20 lb. blast. Records of tests on electric engines of similar size and load show about 84 per cent efficiency between gas cylinders and shaft. The difference between 79 per cent and 84 per cent is 100 h.p. chargeable to air tub friction. The remaining 320 h.p. is the friction loss between gas cylinders and shaft. Therefore the shaft horsepower is 1980−320, or 1660. The friction loss of the tubs is $\frac{100}{1660}$, or 6 per cent, and the mechanical efficiency is 94 per cent. This applies to Slick tubs having a minimum of valve gear. For the average of various types of tubs the mechanical efficiency is taken at 92 per cent. These three efficiencies of the air tub taken together show a net shaft efficiency of about 77 per cent within the range of 15 to 20 lb. blast pressure.

The thermal efficiency at the shaft is about 21 per cent for normal operation at a blast pressure between 15 and 20 lb., which is based on the results obtained from tests on Snow engines. Two series of tests

were made, one at constant pressure of 20 lb. with variable speed, and the other at constant speed of 60 r.p.m. with variable pressure. The former showed a thermal efficiency of 30 per cent referred to the i.h.p. of the gas cylinders at 40 to 65 r.p.m. and the latter showed 29 per cent on the same basis at 10 to 20 lb. blast pressure. These efficiencies referred to the gas cylinder power are equivalent to 24 or 25 per cent thermal efficiency referred to shaft power. These tests were made at higher loads than necessary for blowing two-thirds of the requirement of a blast furnace, at which load 23 and 24 per cent were obtained. Since the results were secured during comparatively short test periods of four hours each and since there is no assurance that they would be duplicated in long periods, 21 per cent thermal efficiency is considered a reasonable figure for the purpose of this discussion.

The net thermal efficiency referred to the theoretical air horsepower is the product of the shaft efficiency and the thermal efficiency at the shaft. This amounts to 16.2 per cent at 15 to 20 lb. blast pressure and varies from 9.5 per cent at 5 lb. pressure to 17.8 per cent at 30 lb. pressure.

Operation Expense. As in the case of cost of installation it has been necessary to estimate the costs of the items making up the total cost of operating the turbo-blower equipment. The basis for estimating the gas-cleaning and boiler-plant expense is the actual cost of similar operation under steel-mill conditions and the expenses applying to the turbo-blowers are based on detailed analysis of the various operating costs of a number of high-grade turbo-electric plants of the same size approximately, which seem to be proper sources of data on account of the similarity of equipments. In the case of gas blowing engines the operation expense is the direct result of comparing the actual costs for several gas blowing plants.

Cost of Operation (without Fuel) of Turbo-Blower Installation for Four 500-Ton Blast Furnaces

	Per Year	Per 1,000,000 Cu. Ft. Delivered (74,845 million)
Gas cleaning	$ 8,500	$0.113
Boiler plant expenses	24,000	0.321
Engineers, wipers, etc	14,600	0.199
Labor and material in repairs and maintenance	10,000	0.134
Lubricants	750	0.010
Tools and miscellaneous supplies	750	0.010
Water	2,500	0.033
Total	$61,100	$0.820

Cost of Operation (without Fuel) of Gas Blowing Engine Installation for Four 500-Ton Blast Furnaces

	Per Year	Per 1,000,000 Cu. Ft. Delivered (74,845 million)
Cost of purification	$17,600	$0.235
Engineers and wipers	21,100	0.282
Labor and material in repairs and maintenance	20,000	0.268
Lubricants	8,200	0.110
Tools and miscellaneous supplies	850	0.010
Water	4,400	0.059
Electric power	350	0.005
Total	$72,500	$0.970

Fixed Charges. Since there are no turbo-blower installations of this particular size and equipment, the cost of installation has been estimated with due consideration of available data on actual costs of material and construction of similar apparatus. In the case of the slow-speed gas blowing engines, the cost is an estimate on a model installation, combining the best features of a number of actual plants and making use of data on the actual expenditures for plants of capacities almost identical with the turbo-blower installation. Although there are at hand no data on the costs of exclusively high-speed gas blowing-engine installations, use has been made of the afore-mentioned data on slow-speed equipments modified to take account of reduced space and present prices of the high-speed equipment.

Cost of Installation and Fixed Charges on a Turbo-Blower Equipment Complete for Four 500-Ton Blast Furnaces

	Cost		Estimated Life, Years	Depreciation at 6 Per Cent Interest	Sinking Fund Per Year	
Gas-cleaning plant		$27,000	15	4.2		$1,130
Boiler house						
Excavating and filling	$1,800		..	...	...	
Foundations	2,520		20	2.8	70	
Building	30,600		20	2.8	860	
Floors, etc.	1,080		20	2.8	30	
Total		36,000		2.8		960

COST OF INSTALLATION AND FIXED CHARGES ON A TURBO-BLOWER EQUIPMENT COMPLETE FOR FOUR 500-TON BLAST FURNACES—*Continued*

	Cost		Estimated Life, Years	Depreciation at 6 Per Cent Interest	Sinking Fund Per Year	
Boilers						
Excavating and filling	1,080		..	...	..	
Foundations	5,400		18	3.3	180	
Boilers complete	100,800		18	3.3	3,320	
Steam pipes	28,800		18	3.3	950	
Feedwater system	18,000		15	4.2	760	
Blow-off pipes, etc.	4,320		15	4.2	180	
Gas flues, burners, etc.	25,200		10	7.2	1,810	
Superheaters	32,400		10	7.2	2,330	
Total		216,000		4.4		9,530
Building for turbo-blower						
Excavating and filling	2,500		..	...	...	
Foundations	4,000		20	2.8	110	
Building	26,000		20	2.8	730	
Floors, etc	8,000		15	4.2	340	
Crane	7,500		15	4.2	320	
Total		48,000		3.1		1,500
Turbo-blowers						
Excavating and filling	5,000		..	...	...	
Foundations	10,000		15	4.2	420	
Turbo-blower erected	320,000		10	7.2	23,000	
Condenser	120,000		10	7.2	8,640	
Piping and sewers	45,000		10	7.2	3,240	
Blast pipes	48,000		18	3.3	1,580	
Total		548,000		6.7		36,880
Total investment		875,000				

Fixed charges	Per Cent	Per Year
Depreciation	5.7	50,000
Interest on investment	6.0	52,500
Taxes	1.5	13,120
Insurance	0.5	4,380
Total	13.7	120,000
Total per 1,000,000 cu. ft. air		1.60

COST OF INSTALLATION AND FIXED CHARGES FOR A SLOW-SPEED GAS BLOWING ENGINE EQUIPMENT COMPLETE FOR FOUR 500-TON BLAST FURNACES

	Cost		Estimated Life, Years	Depreciation at 6 Per Cent Interest	Sinking Fund Per Year	
Gas-cleaning plant						
Excavating and filling....	$2,400		..	...	...	
Foundations.....	9,000		10	7.2	650	
Building.....	7,000		15	4.2	300	
Gas washers (static).....	37,000		15	4.2	1,550	
Gas holder.....	8,200		15	4.2	340	
Machinery (rotary washer)	23,200		10	7.2	1,670	
Piping.....	29,200		15	4.2	1,230	
Total.....		116,000		4.9		5,740
Building for engines						
Excavating and filling...	7,000		...	...	...	
Foundations.....	14,000		20	2.8	390	
Building.....	122,000		20	2.8	3,420	
Floors.....	34,000		15	4.2	1,430	
Crane.....	12,000		15	4.2	500	
Total.....		189,000		3.0		5,740
Gas blowing engines						
Excavating and filling....	14,000		..	...	....	
Foundations.....	60,000		15	4.2	2,520	
Engines.....	1,096,000		10	7.2	78,910	
Piping.....	60,000		10	7.2	4,320	
Blast pipes.....	75,000		18	3.3	2,770	
Total.....		1,305,000		6.8		88,520
Total investment.....		1,610,000				

Fixed charges	Per Cent	Per Year
Depreciation.....	6.2	100,000
Interest.....	6.0	96,600
Taxes.....	1.5	24,150
Insurance.....	0.5	8,050
Total.....	14.2	228,800
Total per 1,000,000 cu. ft. air.....		3.06

COST OF INSTALLATION AND FIXED CHARGES FOR A HIGH-SPEED GAS BLOWING ENGINE EQUIPMENT COMPLETE FOR FOUR 500-TON BLAST FURNACES

	Cost		Estimate Life, Years	Depreciation at 6 per Cent Interest	Sinking Fund Per Year	
Gas-cleaning plant						
Excavating and filling	$2,400		..	...	...	
Foundations	9,000		10	7.2	650	
Building	7,000		15	4.2	300	
Gas washers (static)	37,000		15	4.2	1,550	
Gas holder	8,200		15	4.2	340	
Machinery (rotary washers)	23,200		10	7.2	1,670	
Piping	29,200		15	4.2	1,230	
Total		116,000		4.9		5,740
Building for engines						
Excavating and filling	5,000		..	...	...	
Foundations	10,000		20	2.8	280	
Building	93,500		20	2.8	2,620	
Floors	26,000		15	4.2	1,090	
Crane	9,500		15	4.2	400	
Total		144,000		3.1		4,390
Gas blowing engines						
Excavating and filling	10,000		..	...	....	
Foundations	40,000		15	4.2	1,680	
Engines	842,000		10	7.2	60,620	
Piping	55,000		10	7.2	3,960	
Blast pipes	70,000		18	3.3	2,310	
Total		1,017,000				68,570
Total investment		1,277,000				

	Per Cent	Per Year
Fixed charges		
Depreciation	6.2	78,700
Interest	6.0	76,620
Taxes	1.5	19,155
Insurance	0.5	6,385
Total	14.2	180,860
Total per 1,000,000 cu. ft. air		2.41

Notes on Turbo-Blower Installation. The gas-cleaning plant is composed of static wet cleaners of the tower type with a capacity of 3,300,000 cu. ft. per hr. cleaned to 0.25 grain per cu. ft.

The boiler plant consists of 7200 h.p. of water tube gas-fired boilers with provision for coal firing, superheaters, feedwater system, steam and water piping contained in an all-steel building of the class common in steel plants. The normal requirement of the turbo-blowers including auxiliaries and losses is 6000 b.h.p., the remaining 1200 h.p. being for spare use and cleaning.

The turbo-blower plant includes six turbo-blower units driven by direct-connected high-pressure steam turbines, each having a rated capacity of 35,600 cu. ft. per min. at 15 to 20 lb. blast pressure, there being two units to spare. Each blower has a surface condenser for 40,000 lb. of steam per hour and 28.5 in. vacuum with 70 deg. water. Each condenser requires 7200 gal. of circulating water, pumped by an 18-in. centrifugal pump for each unit. The circulating and hot-well pumps are driven by steam turbines and the air pump by a reciprocating engine. All auxiliaries are located in the basement. The water supply to the pumps is through a 7-ft. conduit, assumed 400 ft. long, and the sewer is of similar size and length. The blast piping includes receivers, valves, cross-connections and all piping to the mains outside the building. The building is 50 ft. by 175 ft. of brick, painted inside, with concrete floor.

Notes on Gas Blowing Engine Installation. The gas-cleaning plant consists of a preliminary and a secondary part, the former comprising three wet scrubbers (tower washers) and the latter three Theisen gas washers operated by electric motors. The mechanical equipment (rotary washers) is installed in a steel building with brick walls, painted inside and provided with all auxiliaries such as pumps, switchboard, wiring, gages, safety appliances, etc. A small gas holder acting as gas-pressure regulator is included, as well as a settling tank to purify the waste water from the gas-cleaning plant and all the piping connecting the various parts of it also provided. The total capacity would be about 25,000 cu. ft. of gas per minute.

The blowing-engine house is a steel frame building with brick walls, painted inside, large windows, ventilation, a skylight, reinforced concrete floors, a spacious basement 10 ft. high, 30-ton electric traveling crane with main and auxiliary hoist, safety and sanitary appliances, electric lighting, office, etc. The width of the building for slow-speed blowing engines is about 105 ft. between the center line of columns and it is about 90 ft. for high-speed gas blowing engines.

The gas blowing-engine equipment consists in the case of slow-speed gas blowing engines of eight double-acting, twin-tandem, gas engines about 42 in. by 60 in. stroke, operating blowing tubs of about 72 in. in diameter at a normal speed of 50 to 60 r.p.m. The alternative case of high-speed engine equipment comprises eight double-acting, twin-tandem, gas blowing engines; about 38 in. in diameter of gas cylinders; 64 in. in diameter of blowing tubs, 50 in. stroke, operating normally at about 85 to 90 r.p.m.

Depreciation is understood as the amount of money set aside yearly, which compounded at an accepted rate of interest would equal the original investment at such a time as replacement is necessitated through obsolescence, age or inadequacy.

From these figures it will be seen that while a turbo-blower installation under certain conditions is more economical from a commercial standpoint than an installation of slow-speed gas blowing engines, modern high-speed gas blowing engines, as advocated by Mr. Trinks[1] and as used in German blast furnaces and steel plants, offer marked advantages over the turbo-blower as a financial investment.

The present tendency towards high-speed machinery is sure to extend its influence to the gas engine and particularly the gas blowing engine. The only difficulty which so far has stood in the way in this country of the development of high-speed gas blowing engines is the lack of a blowing tub which would operate successfully on high speeds. This side of the proposition has been very ably and fully dealt with in Mr. Trinks' paper.[1]

A recent trip through Europe has revealed the fact that in spite of the development of the turbo-blower by several manufacturing concerns in Germany, the gas blowing engine is more than holding its own, due to the high price of coal, since certain turbo-blower installations did not give as satisfactory results as had been claimed for this type of blowing engine.

Looking at the proposition merely from a short-range commercial standpoint, it must be conceded that for isolated blast-furnace plants where no use can be made of the excess gas produced by the blast furnaces, the turbo-blower is unquestionably very desirable, in fact, unsurpassed as an ideal apparatus for blowing smaller furnaces, and it is likely to prevent very successfully the introduction of gas blowing engines in such cases, except when blast-furnace proprietors in a district reach such an agreement as exists in several instances in

[1] Reciprocating Blast-Furnace Blowing Engines, Trans. Am. Soc. M. E., vol. 33, p. 429.

Germany, for the purpose of producing cheap electric power for sale to communities and industrial establishments of all kinds located in the neighborhood of these isolated plants.

Thus, even in isolated blast-furnace plants very careful investigations should be made before the gas engine is discarded as a possible competitor for steam equipment, since a lucrative field for disposal of surplus electric power may be thrown open by the organization of such syndicates of otherwise competing blast-furnace and steel works. They would have the additional advantage that each constituent company could obtain power from the common system at low cost in case their own power production is curtailed by accidents to their machinery or by reduced output of their furnaces.

In spite of all computations endeavoring to prove that the steam turbine or turbine-driven turbo-blower are commercially superior on account of the low cost of installation and operation, it is impossible to get around the fact that the surplus gas of blast furnaces when converted into power in internal-combustion engines, yields over twice the amount of energy possible when the conversion takes place in the roundabout way of steam boiler and reciprocating engine or steam turbine. To emphasize this statement it needs only to be pointed out that of the total gas production of four 500-ton blast furnaces of about 13,000,000 cu. ft. per hr., there remains available for purposes outside the blast-furnace requirements themselves, 3,750,000 cu. ft. per hr. in the case of the turbo-blower installation, while this surplus amounts to 5,670,000 cu. ft. per hr. when gas blowing engines are used. In the former case an electric station of about 10,000 kw. capacity, equipped with steam turbines, could be operated without burning additional coal under boilers, whereas in the latter case a gas-engine central station of about 30,000 kw. capacity could be installed and operated.

In view of the natural tendency towards reducing the cost of installation and operation of power plants and of the constant increase in the price of coal, it is very clear that the internal-combustion engine is bound to gain in value as a prime mover in the future. The smoke nuisance, the difficulties of coal shortage, car famine, etc., in the case of steam-power installations are mentioned only incidentally in their influence upon the probable development of gas power.

Recently means have been found to increase still further the efficiency of gas-power plants by recovering the waste heat formerly lost in cooling water and exhaust. Experiences in a large Belgian blast-furnace and steel plant have proved that about 13 per cent more power can be obtained from the same input of heat units by harness-

ing in the form of low-pressure steam a large portion of the waste heat of gas engines.

In power plants of the future, the installation of waste heat recovery appliances will permit of obtaining continuous high thermal efficiency of the gas engines by maintaining uniform high load on these engines, while peak loads will be carried by low-pressure, steam turbo-generators.

The suggestion offered by Mr. Johnson in regard to a combination of low-pressure steam turbo-blowers and reciprocating steam-blowing engines, can be considerably improved and elaborated upon, since it would seem much more economical to install in a new blast-furnace plant several large turbo-blowers, operated by low-pressure steam turbines, running on steam generated by the waste heat recovered from cooling water and exhaust of gas electric and gas blowing engines. These turbo-blowers would deliver the total amount of air necessary for all furnaces at low pressure into an air-tight tunnel or header, from which a number of gas blowing engines would be supplied. The latter would compress the blast in a second stage, as it were, to regular blast pressure.

With such a combination the fuel cost for obtaining low-pressure blast would be practically nothing and the cost of operation and fixed charges moderate, on account of the rotary type and reduced cost of installation of the first-stage machinery. Moreover, the first cost and operating expenses as well as fuel cost for the gas blowing engines themselves would be reduced, since a smaller number of smaller gas blowing engines would suffice. The volumetric efficiency of the blowing tubs would be practically 100 per cent, since the air delivered by the turbo-blowers could easily be cooled by means of pipe coils arranged in the air-tight tunnel, through which the engine cooling water would circulate before entering the gas-engine water jackets.

The safety of blast-furnace operation would suffer in no way, since in case of shutdowns of the turbo-blowers, the gas blowing engines would insure delivery of a reduced quantity of air; while in the event of trouble with the gas engines, large quantities of low-pressure air would be available to keep the gas out of the cold blast pipes.

Since conservative estimates show that at least 12 per cent of the power produced in gas engines is available for low-pressure steam turbines in the waste heat, 25 per cent and more of the power necessary for blowing the furnaces could be secured by the utilization of the waste heat from gas blowing engine and gas-electric engine installations.

THE AUTHOR. The question of efficiency and accuracy of control of the rate of flow of air into the furnace, which is one of the vital questions to be considered in connection with turbo-compressors, consists in reality of two parts, (*a*) a knowledge of the exact rate of flow at any particular time, and (*b*) the accuracy with which this rate of flow can be changed. In both cases the turbo-compressor stands alone, and by virtue of the great accuracy with which the rate of flow can be determined, and controlled and varied through as slight or as great ranges as may be desired, will give unprecedented control over the variations of the furnace to the furnace man, and enable him to produce more uniform quality of iron under greater quantity from any given furnace.

The variation per stroke in volume discharged from any given reciprocating compressor when blowing at pressures of 10 or 20 lb. (both of which are common in practice), due to the variations in the quantity of air required to fill the clearance space at these different pressures, is far greater than the variations due to the inaccuracy of the calibration of the constant volume governor scale beam on the centrifugal compressors. In addition the reciprocating blowing engine suffers from variations due to leakages, pulsations in the blast main, and other causes which, when summed up, render uncertain the exact quantity of air flowing into the furnace. Uncertainties do not exist when the centrifugal compressor is used, for the blast-furnace operator is furnished with a means of measuring the air accurately and controlling it with any necessary degree of precision.

The method proposed by Mr. Dreyfus of blowing air to a constant pressure in a turbine-driven compressor and taking constant pressure into a reciprocating blowing engine is not one made necessary by the considerations set forth. The turbo-compressor can be designed with a reasonable number of stages to give pressures required in the normal operation of the blast furnace, and by a very moderate increase in speed it can be made to give, with good efficiency, the maximum pressures required in such service. Therefore it is not necessary to use this type of blowing engine as a booster. Simplification and accuracy of operation are obtained in the highest degree consistent with maximum economy by taking the air from the turbo-compressor directly into the furnace.

The method of operation where the turbo-compressor is used as a booster does not realize the full benefits of the accurate control and uniform flow of air afforded by the turbo-compressor, and it is there-

fore objectionable. It is, however, extremely economical to use steam from reciprocating blowing engines in a mixed pressure turbine driving a turbo-compressor, and several such installations are now in contemplation. In such cases, the best arrangement is for the reciprocating blowing engines to blow directly into one furnace, and the turbo-compressors into another, allowing each apparatus to be governed in the ordinary manner by its own governing mechanism.

The use of dry blast in a turbo-compressor results in increased efficiency, since the air is delivered to the turbo-compressor at a greater density than where it is taken from the atmosphere direct, and this greater density results in the production of the pressure at a lower speed and less consumption of steam.

The description of the centrifugal air compressor developed by the Westinghouse Machine Company and the high efficiencies claimed for this machine are extremely interesting. These latter would be gratifying if realized in practice, but tests of the Great Falls apparatus which have come under the knowledge of the writer indicate that the measurements of efficiency are made on an erroneous basis, and that such machines not provided with discharge vanes, but with rather imperfect types of impellers, are not able to give the efficiency realized by the apparatus which form the subject of the author's paper.

Mr. de Laval called attention to the fact that it is not necessary to use six stages in series for pressures as low as 10 to 20 lb. This is undoubtedly true where higher speed turbines are available for driving such apparatus. At the time the Empire machine was designed such was not the case. Machines for these pressures are now provided with only three stages and attain a higher efficiency because of the higher speed and the smaller losses resulting. Under these circumstances, it is not desirable to reduce the diameter of the second or third impeller as advocated by Mr. de Laval, but only in cases where the pressures to be realized are considerably higher.

The great differences between the Rateau multi-stage compressors and those of the type described by the author reside in the method of balancing end thrust, by Rateau in separate impellers and in the other type by double inlet impellers, and in the governing mechanism.

The arrangement of impellers used by Rateau results in "pumping" of air back and forth between one and the other at light loads. This pumping effect does not exist in the machine described.

The radical difference in the governing mechanism lies in the General Electric governing mechanism being placed on the inlet, and, therefore, subject to far less variations in density, pressure and temperature of air than where the governor is on the discharge.

Professor Trinks mentioned that many of the turbo-blowers abroad were not in operation. The author believes this to be due to the fact that the method of governing such machines is not sufficiently accurate to cause good results, whereas the accuracy of the apparatus described in this paper leaves nothing to be desired.

Mr. de Laval referred to an automatic outlet valve used to prevent detrimental pulsation of air at partial capacities. The same remarks that have been made in connection with the air governor apply to this apparatus. After extensive experiments the author believes it is much better to control the air pulsations by apparatus on the inlet than on the discharge.

It is gratifying to see that Messrs. Freyn and Bacon apparently agree that for installations involving the blowing of less than four furnaces, the turbo-blower is superior to the gas engine, since they do not attempt to study installations involving a smaller number of units. As regards their figures for plants involving blowing four furnaces or more, there are a number of serious objections to be made. In addition, the author feels that if Messrs. Freyn and Bacon had given full credit to the increased value of the output due to the more accurate governing of the turbo-compressor, and the consequently increased quality of the product and its increased amount due to the more uniform operation of the furnace resulting from the employment of the turbo-compressor with this method of governing, their conclusions would have been modified even in the case of the plants they discuss. It is difficult perhaps for furnace men to realize the value of this feature or to give it proper credit.

Referring to the figures for the installation and operating expenses of various plants, it will be noted that the assumption is made that each gas engine shall have the capacity to blow two-thirds of a furnace. Such a method of operation of furnace blowing units is extremely undesirable, since it leads to the imposition of tremendous strains upon the blowing engine, requiring the outlay of a considerably larger investment cost than has been used by these authors in their tables, or resulting in a much larger cost of repair and maintenance than they have used. Nothing approaching the best results in the operation of a blast furnace can be obtained by such a method of operation. The author knows of two cases where in the use of such

a method serious difficulties and abnormal expenses for repairs to blowing engines resulted.

The adequacy of the number of units specified by these authors is questioned in the case of gas engines to provide the necessary amount of spares that the operation of the group of four furnaces can be maintained at full capacity at all times when the condition of the furnace renders such operation possible. Information from all sources as to the continuity of operation of gas engines for blowing blast furnaces confirms the opinion that instead of eight units as specified by Messrs. Freyn and Bacon, ten units would be none too many. If such is the case, the figures for the cost of installation, interest charges, repairs, operating costs, etc., must be considerably raised against the gas blowing engine with turbo-compressors for this purpose. In this case also, if sufficient first cost was put into the blowing engines to secure necessary reliability and to render the above number of units sufficient, the investment cost would have to be largely increased.

Messrs. Freyn and Bacon have assumed that the entire plant of four furnaces would operate continuously at the rated capacity of 500 tons per day. The author believes that such an installation operating continuously for any considerable length of time at the rated capacity has never existed. Messrs. Freyn and Bacon state: "In case of reduced output, or of partial shutdown, due to dull business, relining furnaces, etc., the reduction in the item of fuel expense would be about twice as great with turbo-blowers as with the gas engines, while the item of fixed charges would continue at the same rate, resulting in a greater net decrease in the total expense for the turbo-blowers than for the gas engines." That is to say, if ordinary instead of theoretical operating conditions had been assumed, the case for the turbo-blowers would be substantially improved.

The vital element in the argument of Messrs. Freyn and Bacon for the maintenance of the gas engine is the question of its efficiency of operation. It is to be regretted that it is not stated how the efficiency of the gas engine or reciprocating blowing tubs was ascertained. It is extremely difficult, if not impossible, to measure correctly by any means in common use the discharge of air from a reciprocating blowing apparatus, owing to the variations in pressure met with in the blast mains. This is due to the fact that the air is discharged from the blowing tub during only a fraction of the stroke, causing a pressure wave to proceed through the blast main at every stroke. It may be that the blast main in this particular case was

so long that these fluctuations in pressure were smoothed out and became negligible; but it would be very interesting to have this point cleared up. If the quantity of air discharged was measured by the indicator card it must be obvious that large errors are likely to exist, and this would seriously affect the computations of efficiency. For instance, if a leak exists in the discharge valves the indicator cards will show a considerable amount of work performed while the air delivery is not in proportion to the amount of work so shown. In comparing turbo-compressors with reciprocating blowing apparatus of any type whatever, this fact should be kept clearly in mind. The quantities discharged by the turbo-compressor are accurately measured within 2 or 3 per cent. The quantity of air discharged by a reciprocating machine is not capable of accurate measurements in the ordinary installation, and figures of volumetric displacement as compared with actual delivery are admittedly at variance by considerable and variable percentages.

Messrs. Freyn and Bacon, in discussing the efficiency of turbo-compressors, assume a figure of $62\frac{1}{2}$ per cent for boiler and furnace efficiency, which is perhaps justifiable if the plant is to be operated in the usual blast-furnace manner. It is practicable by the installation of superheaters and fuel economizers to secure with proper supervision 70 per cent efficiency. Of course if this per cent were used in computations, the figures for the turbo-blower installation would be greatly improved.

In discussing the turbo-blower, Messrs. Freyn and Bacon assume an operating steam gage pressure of 135 lb. There is no reason why a modern plant involving the installation of the best apparatus developed for the purpose could not operate at a pressure of 175 lb., but certainly any figures for cost of operation of turbo-blowers ought to be based on a pressure no lower than 150 lb. Furthermore, Messrs. Freyn and Bacon increase the steam consumption from 11.9 lb., which they derive from a consideration of factors, including this low steam pressure, and the use of efficiency of 70 per cent, which is lower than the figure they themselves give for the turbo-compressor ($72\frac{1}{2}$ per cent), and which is correct, to 12.5 lb., or an increase of 5 per cent. Since they have decreased the efficiency as above by $8\frac{1}{8}$ per cent, this makes a total decrease of efficiency of the turbo-blower, to allow for variable conditions, of $8\frac{1}{2}$ per cent. All the curves and tests on turbo-compressors made by the author indicate that this figure is very much too high. The figure given by Messrs Freyn and Bacon for the performance of turbo-blowers is therefore unnecessarily conservative.

Although the loss by variation of conditions of air pressure and volume is greater in the gas engine than it is in the steam turbine, no allowances for the effect of these variations have been made by Messrs. Freyn and Bacon in discussing the efficiency of the gas engine, while they have been made when discussing the turbo-compressors. If these allowances are eliminated and 70 per cent is used for furnace efficiency the thermal efficiency of the turbo-blower installation complete would be 11.6 per cent at brake horsepower, or 8.1 per cent at horsepower of adiabatic compression, instead of 9.46 and 6.62 given by these writers. These figures would modify considerably the comparison with the gas engine.

The thermal efficiency of 21 per cent at the shaft, which Messrs. Freyn and Bacon give for the gas blowing engine, should be reduced to at least 19 per cent, if the results of the variations met with in practice are taken into account. In all probability, under all the variable conditions of blast-furnace operation this efficiency would be considerably less than 19 per cent. This would also reduce the thermal efficiency at theoretical air horsepower, given by these writers as 16.2, to not more than 14.6.

In regard to the cost of operation and maintenance, it is of course very difficult to get any figures on the actual cost of operation of installations of either type of blowing apparatus, but from available information the author is inclined to question seriously the adequacy of the item of "labor and material in repairs and maintenance of the gas blowing engine at $20,000 per annum," as being in any way sufficient to cover the cost of this item for an installation of 8 or 10 gas blowing engines.

Referring to the statement of these authors, "In spite of all computations endeavoring to prove that the steam turbine or turbine-driven turbo-blower are commercially superior on account of the low cost of installation and operation, it is impossible to get around the fact that the surplus gas of blast furnaces when converted into power in internal-combustion engines yields over twice the amount of energy possible when the conversion takes place in the roundabout way of steam boiler and reciprocating engine, or steam turbine," it should be borne in mind that the output of gas from a blast furnace is variable and irregular in amount. The author believes that Messrs. Freyn and Bacon do not take sufficient account of the irregularity of operation of plants depending on this gas for their motive power, where such plants cannot be provided with additional sources of heat, for instance, by hand-fired boilers. It is true that a gas

engine can use only gas, and if the gas supply is not sufficient, the gas engine must give out a reduced output, or shut down. In the case of a steam plant containing boilers, if the gas supply is insufficient, the capacity can be kept up by hand-firing of coal. In the case of a gas engine, unless an expensive plant of producers is installed, a deficiency in output will be experienced under such conditions. For these reasons, speculations as to the value of the additional gas set free by the use of a more economical blowing engine must be considered in the light of this irregularity, and additional apparatus consisting of steam engines or steam turbines, provided with the necessary boilers, etc., or a plant of gas producers must be installed to supplement the gas-engine plant if regularity of operation is to be expected and regularity of output ensured. Furthermore, it is difficult properly to govern a gas engine to insure the necessary regularity and frequency of voltage, and it has been found necessary in some cases to install one or more steam turbines in connection with a gas-engine electric plant taking its gas supply from blast furnaces, in order to provide the necessary regulation. The cost of such installations ought to be added to the cost of investment in the gas-engine plant in order to bring the figures up to a fair basis of comparison.

It should also be borne in mind that the present direction of progress in the operation of blast furnaces is towards a reduction in the amount of the fuel burden. For many years blast-furnaces have been using about twice the theoretical quantity of fuel. It is not impossible that in the near future this figure will be greatly reduced, resulting (*a*), in a decrease in the quantity of gas, and (*b*), in a decrease in its quality. This deficiency will have to be made up by hand-fired boilers or coal-fired producers. Considering the comparison between steam and gas units under these conditions it should be borne in mind that the gas producer is an inefficient piece of apparatus and this, with the low combined efficiency of all the conversion processes involved, will militate against it in such a comparison.

Messrs. Freyn and Bacon devote considerable space to the discussion of a combination of a gas-blowing installation and gas-electric station to which are attached several large turbo-blowers operated by low-pressure steam turbines running on steam generated by the waste heat recovered from cooling water and exhaust. The undesirable features of the tandem operation of turbo-blowers and gas blowing engines which these writers advocate have already been dealt

with. The author furthermore believes that they greatly overestimate the quantity of such available heat, which under the most favorable conditions would amount to not more than 10 per cent. Moreover, he cannot see how under such a combination of conditions such as the writers set forth, a volumetric efficiency of anything like 100 per cent could be secured. He is ready to admit, however, that the leakages would be somewhat reduced by reason of the lower range of pressures dealt with by the reciprocating compressor, but the main causes for the low volumetric efficiency would still remain present.

In conclusion, taking figures obtainable by the average purchaser, made up on the basis used by Messrs. Freyn and Bacon, and giving full credit to the turbo-blower for all improvements made in the operation of the furnace, both as regards quantity and quality of output for the average plant, the gas engine will be found to have a very limited field of application, and only in the very largest plants and under the most favorable conditions in regard to fuel cost will it be found desirable to install such apparatus.

No. 1321

RECIPROCATING BLAST-FURNACE BLOWING ENGINES

By W. Trinks, Pittsburgh, Pa.

Member of the Society

In the last 20 years American blowing-engine practice has assumed rather set forms. The growing size of furnaces has increased blast pressures to a point where the heat of compression has made the use of felt, canvas, or leather valves impossible. Certain forms of valves and engine types have been developed which have dominated the market for a number of years, and their operation furnishes today the blast for more than 90 per cent of the pig-iron capacity of the United States. But a few years ago the contentedness of American builders and users of blowing engines was rudely shattered by a double European invasion: the gas engine and the turbo-blower.

2 The gas engine, although much more economical of fuel than the steam engine, is more expensive with regard to first cost. To reduce the cost per horsepower, high piston speed must be employed; thus the piston speed has been increased from the 300 ft. per min., heretofore considered standard in steam-driven blowers, to 600 ft. per min. in modern American gas-driven blowers. In Europe reciprocating blowers run at piston speeds of 750 ft. per min.; the gas engine for generation of power runs today at piston speeds very close to 1000 ft. per min.

3 The turbo-blower appeared on the market a few years ago first on the European continent, where its progress was completely checked by the price of coal. The steam turbine could not compete with the low fuel consumption of the gas engine. Recently the turbo-blower has also been introduced in this country. Here the lower price of coal and the higher price of operating labor will favor its introduction in a few districts, but the old law that progress in one line of business invariably begets progress in a competing line is true also in this case, and thus we find builders of reciprocating

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

engines busy making improvements and raising the standard of their product.

4 The understanding of the reasons why the standard types of American blowing engines are so successful at medium speeds and what their shortcomings are at high speeds will be much facilitated by a short study of valve motion and of throttling losses through valves.

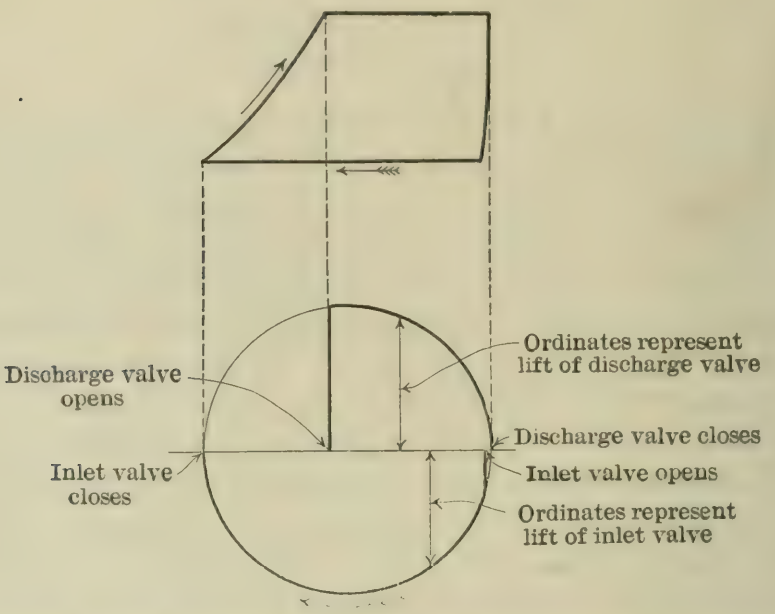

Discharge valve opens
Ordinates represent lift of discharge valve
Discharge valve closes
Inlet valve closes
Inlet valve opens
Ordinates represent lift of inlet valve

FIG. 1 IDEAL VALVE MOTION OF PUMPS OR COMPRESSORS ON DISPLACEMENT BASIS

5 It is proved below that high velocity through valves is harmful; the tendency is therefore to keep the velocity at a fairly constant low value; and since the piston of an engine has very nearly harmonic motion, it follows that the valve should also have harmonic motion, if a constant and small pressure drop is desired. On a displacement basis harmonic motion becomes a circle. Fig. 1 shows the valve lift on a displacement basis. The inlet valve is open practically throughout the stroke; the outlet valve should pop open near the middle of the stroke and close at the end.

6 Fig. 2 shows this same ideal diagram on a time basis. It will be noted that the curves intersect the base line at an angle, indicating that if these ideal motion curves are realized the valve will strike a blow in seating. The velocity of striking depends upon the lift of the valve and upon the time of one revolution, which

means that high-lift valves, while successful for low rotative speeds, become impractical for high rotative speeds.

7 Consciously or unconsciously, engineers have striven to design valves and valve gears for blowing engines so that the above valve motion curves are approximated. Two circumstances interfere with attaining the ideal; one is the quantity of air under a lift valve, which lessens the discharge through the valve during its lift and increases the discharge during the closing period; the other is the mass of the valve which has baffled designing engineers quite

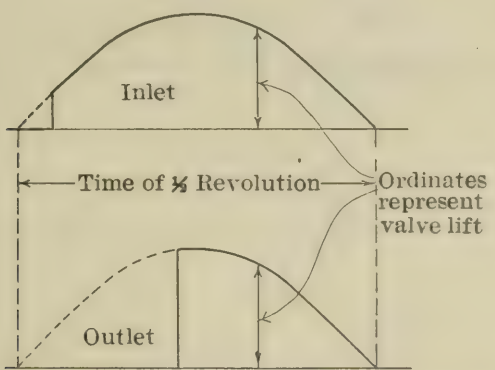


FIG. 2 IDEAL VALVE MOTION OF PUMPS OR COMPRESSORS ON TIME BASIS

often. Twelve years ago, when the author's experience was mainly theoretical, he published formulae for designing valves and springs in such a way that the pressure difference on the two sides of the valve and the impact of the air flowing through the valve would be nicely balanced by the spring and that the ideal valve lift curve could be attained. He was very much surprised by valve vibrations, when the formulae were put to a test. As a pendulum swings about its position of equilibrium, so a valve will vibrate about its ever changing position of equilibrium, if it is designed to float between the air pressure and current on one side and the spring on the other side. A typical (diagrammatic) valve lift curve is shown in Fig. 3. A diagrammatic or made-up curve was preferred to a curve taken from a particular engine, because an endless variety of vibrations is possible, depending upon the mass and shape of the valve and upon the spring loading. It should be understood that the fluttering of the valve occurs to this extent only if no means have been provided for damping it, and that the heavier the valve the more energy the damping consumes. In Fig. 3 the valve is shown to close late.

Practically all automatic lift valves close late, depending upon rotative speed of engine, lift of valve and spring loading. The lower the rotative speed and the valve lift and the greater the spring load closing the valve, the nearer the valve is to the seat in the dead center position of the crank. Tests and calculations show that in ordinary American blowing-engine practice the valves close so near the dead center that for all practical purposes they may be considered as closing "on time" and without the injurious effects of late closing, namely, slipping back of air and hammering of valve.

8 The behavior of the outlet valves is similar to that of the inlet valve with two exceptions: First, the valve has to open when the piston moves fastest; second, the pressure difference on both sides of the valve increases rapidly at the dead center. In Fig. 4 the

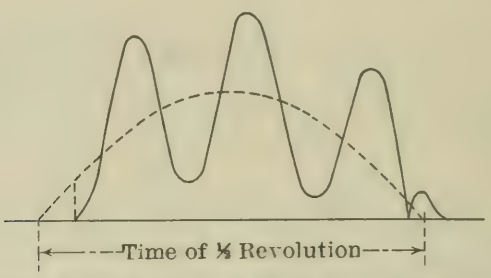


FIG. 3 ACTUAL MOTION OF HEAVY, NON-CUSHIONED INLET VALVES (TIME BASIS)

deal and the actual valve-lift curves are shown in connection with an indicator card on a displacement basis. The slower the rotative speed of the engine, the lighter the valve and the smaller its lift, the more perfect is the approximation to the ideal lift curve. High rotative speeds, heavy valves and small areas uncovered by the valve for a given lift cause the pressure in the cylinder to rise considerably over that existing in the blast space and store up considerable amounts of kinetic energy in the valve which must be taken care of by a cushioning device; otherwise fluttering or hammering results. As already stated, the pressure drops rapidly under the valve immediately after the crank has passed the dead center, and particular care must be taken to have the valve close promptly in order to avoid hammering.

9 Throttling loss through valves involves two factors, loss of velocity head and surface friction. The vast majority of valves are so designed that surface friction, such as occurs in long pipes or ducts, is practically absent. For this reason, losses due to velocity

head only will be investigated here. Fig. 5 gives the throttling loss for various piston speeds and for various ratios of valve area to piston area. In this chart, valve area does not mean the so-called free valve area which is a rather imaginary or conventional quantity, but it means the area actually offered to the air-flow at the narrowest place of the valve. It is assumed that the valve has harmonic motion and that the coefficient of discharge is 70 per cent. For a number of valves this latter figure was found to agree most closely with tests.

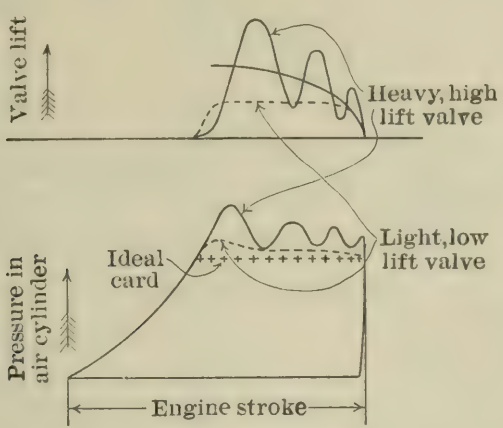
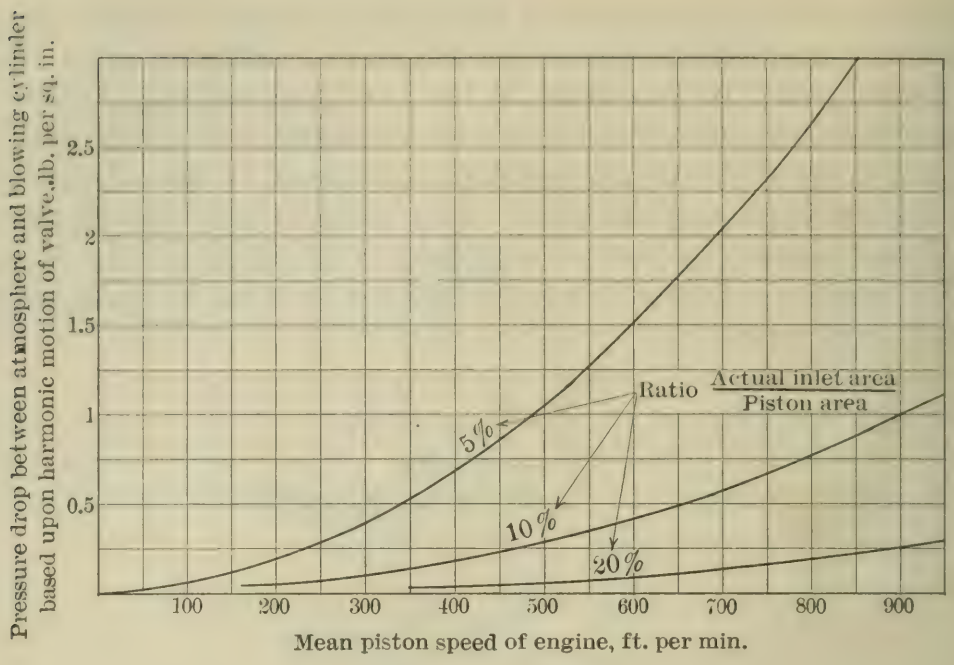


FIG. 4 ACTUAL MOTION OF OUTLET VALVES (DISPLACEMENT BASIS)

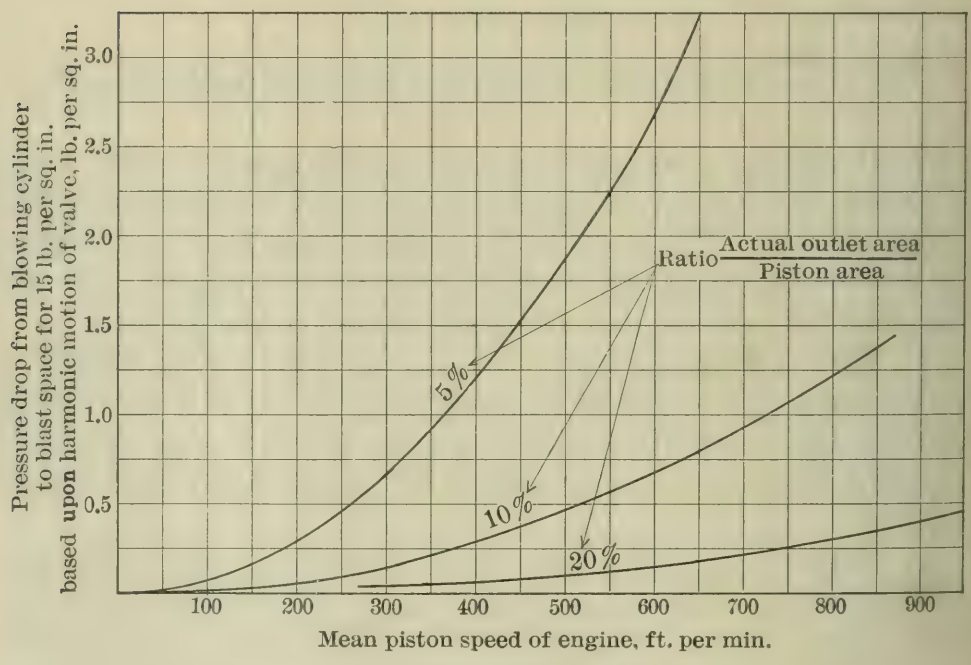
10 It will be observed that for a piston speed of 300 ft. per min., which up to a few years ago was standard in this country, and for inlet areas of 12 per cent, outlet areas of 9 per cent, the pressure drop due to throttling is a negligible quantity compared with the mean effective pressure (say 11 lb. per sq. in.) of the air card.

11 With a stroke of 60 in., so common for blowing engines, a piston speed of 300 ft. per min. means 30 r.p.m. At this low rotative speed the spring loading required to close the valves on time is so small that the inlet and outlet areas of 12 and 9 per cent can be realized.

12 Further penetration into theory would be out of place here. A better idea of the evolution of the modern blast-furnace blowing engine may be gained by the study of standard types. Fig. 6 illustrates a design which for the last 15 years has been very popular in the United States. The inlet valve is mechanically operated throughout and is a balanced piston valve. Mechanical operation of valves has been practised for over 20 years and avoids many of the troubles incidental to an improperly designed automatic valve. No fluttering



(*a*) INLET VALVES



(*b*) DISCHARGE VALVES

FIG. 5 PRESSURE LOSS IN BLOWING-ENGINE VALVES DUE TO VELOCITY HEAD

or chattering can occur, a full port opening may be obtained and the valve may be made to open and close even more rapidly than is required by the ideal sine curve. The breaking of a mechanically operated slide valve is also highly improbable and great reliability therefore results. The outlet valve is a cup, opening automatically by air pressure and closing by mechanical means. The cup cushions

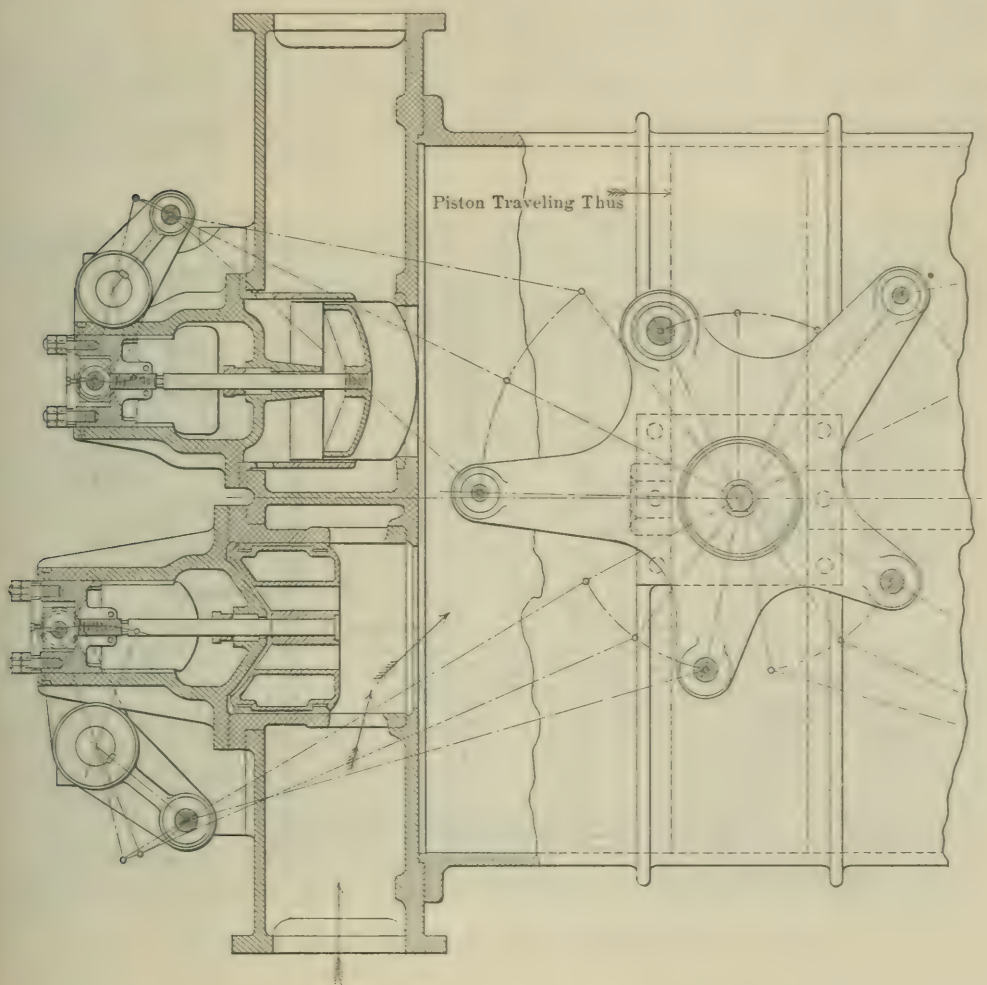


FIG. 6 VALVE GEAR BUILT BY ALLIS COMPANY

without springs against the movable plug, so that fluttering or chattering is impossible; it is fairly light and at rated speed opens with small throttling loss. If the valve gear is properly adjusted, the outlet valve closes at the right time, so that hammering, due to late closing, cannot occur. The number of valves to be looked after is small because a few large, light-lift valves are employed, usually two inlet valves and two or three outlet valves. The valve

gear pins are made so large that very little wear can occur. This precaution keeps the valve gear in correct adjustment.

13 The principles underlying this design are very sound, and its success is well deserved. With engines of 60-in. stroke it is at its best with speeds between 30 and 40 r.p.m. The builders do not recommend it for speeds exceeding 50 r.p.m. At that speed the pressure in the cylinder at the end of the suction stroke is ½ lb. per sq. in. below that of the atmosphere.

14 A very similar valve gear is illustrated in Fig. 7. Here the inlet valve is double-ported for the purpose of quick opening and closing. The outlet valve cushions in opening against a station-

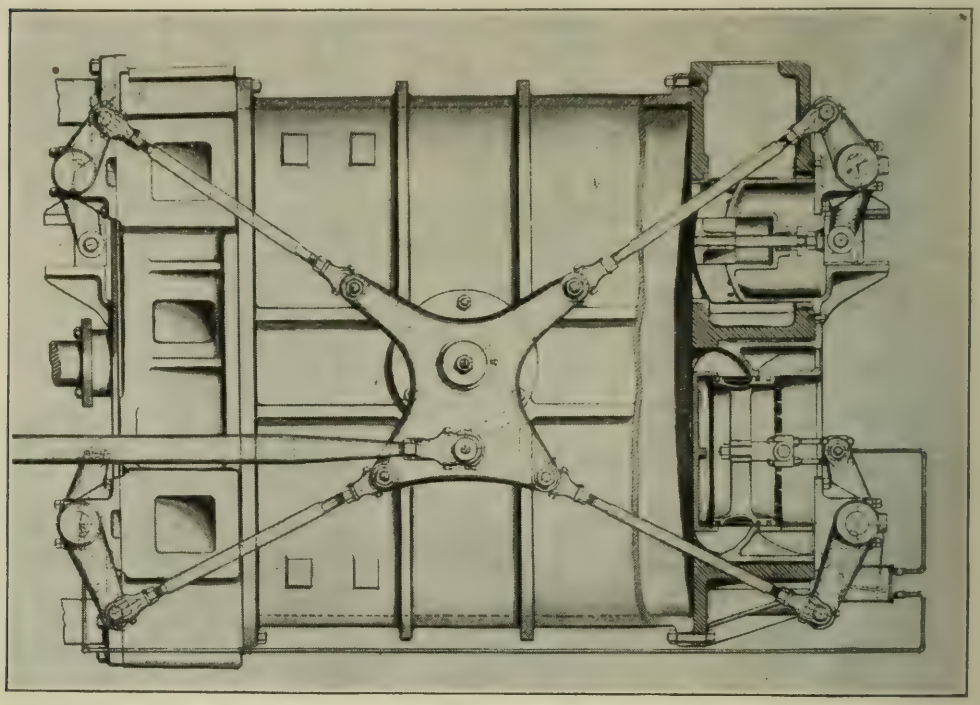


FIG. 7 VALVE GEAR BUILT BY WM. TOD COMPANY

ary plunger and is mechanically closed by a central pusher. In design it is similar to Fig. 6.

15 The fact that two valve gears of such close resemblance, and built by competing firms, could have been patented independently indicates that the valve gear must have good features. It would also seem probable that it was originated in one place and copied in another. As a matter of fact, the outlet valve originated in one concern, the inlet valve in another, and by mutual borrowing two good valve gears were produced.

16 These two valve gears as well as others of standard American design employ mechanically operated, or positively driven, inlet valves. As long as the clearance volume of the engine is small, mechanical operation of the inlet valve is scientifically correct, because the opening and closing points of the valve (Fig. 8) remain practically fixed in spite of variations of blast pressure. Matters are quite different with the outlet valve. Its correct opening point varies with the blast pressure, and losses occur, as indicated in Fig. 8, if the valve opens at a fixed point and if the blast pressure differs from the one for which the engine was designed. Julian Kennedy, of Pittsburgh, reasoned, however, that the gain in reliability by the use of mechanically operated outlet valves would easily offset these

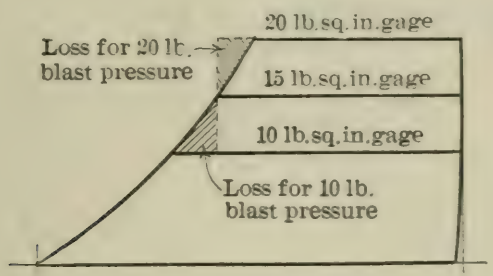


FIG. 8 LOSS CAUSED BY MECHANICALLY OPERATED OUTLET VALVES

losses and that the "work on the engine, due to advance opening of the valve, is probably not in excess of the work required in the case of a poppet valve due to the crowding of the pressure above the receiver pressure due to the opening of the valve." Fig. 9 shows his design and Fig. 10 indicator cards taken from an engine of this type. It will be noted that inlet and outlet valves are balanced piston valves. The piston which is not shown in Fig. 9 is scalloped out to fit around the valves with a view to reducing clearance volume. The outlet valve opens when a pressure of $7\frac{1}{2}$ lb. has been reached in the cylinder. From that point to the reaching of the blast pressure (Fig. 10) the indicator card is a combination of compression and air discharge from the receiver back into the cylinder. An inspection of the indicator card with regard to smallness of lost work shows that Mr. Kennedy is right. The engines built on this principle are outwardly quite successful, and no repairs have been necessary on engines now in operation for over seven years.

17 It is somewhat surprising that a valve gear which positively opens large areas, avoids valve fluttering and prevents slip by timely

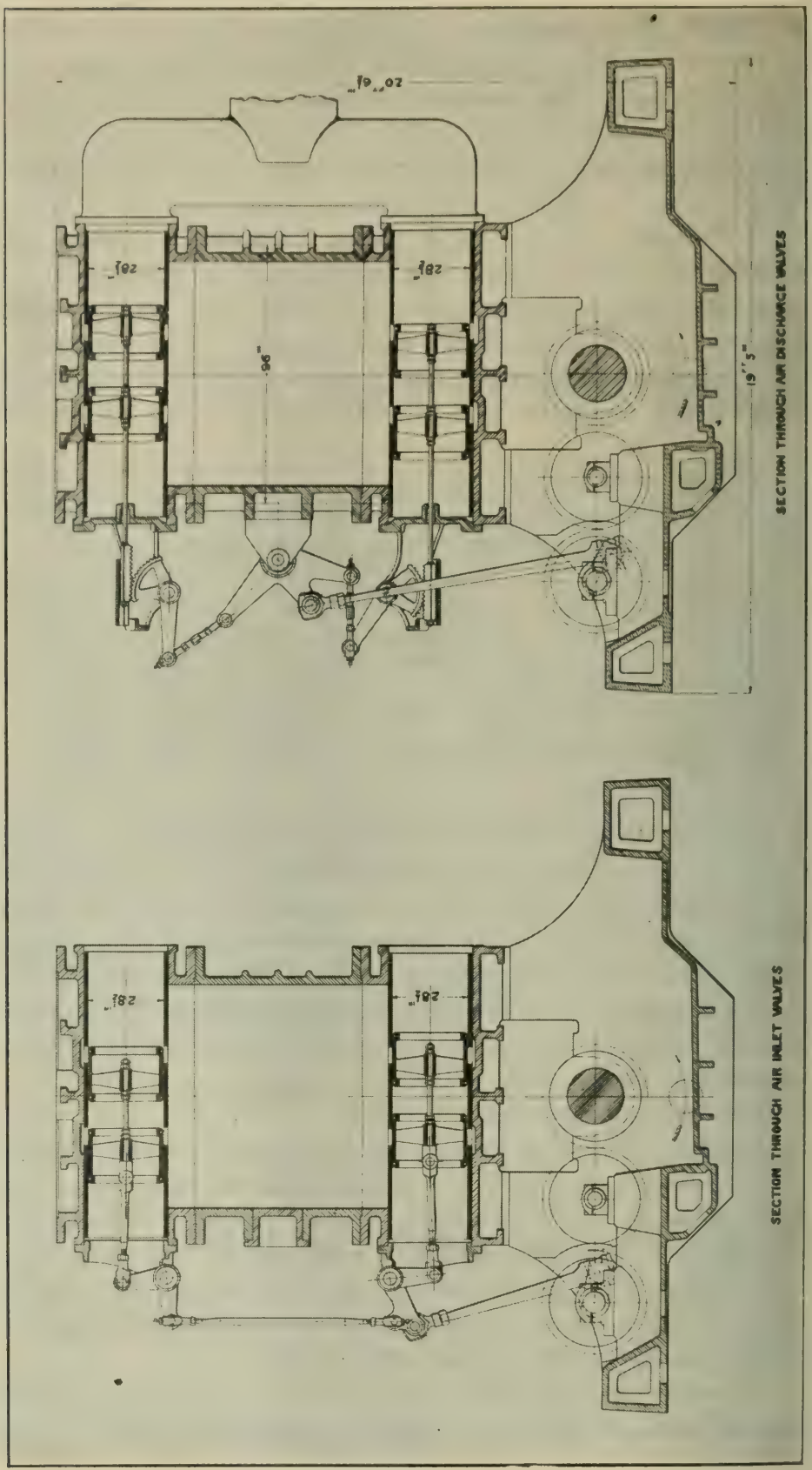
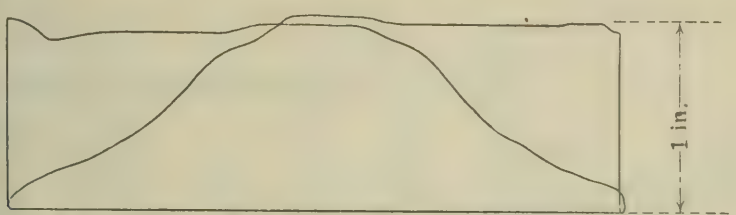


FIG. 9 CYLINDER WITH MECHANICALLY OPERATED INLET AND OUTLET VALVES (DESIGN OF JULIAN KENNEDY)

closing of valves should have been limited to less than two dozen engines. The author believes that the following reasons may have contributed in preventing a more general adoption of the system: (*a*) the large number of pins, rods and levers create the impression of complication; (*b*) large relief valves must be provided to prevent wrecking of the engine in case of accident to the valve gear; and (*c*) the slower the speed of the engine, the greater the work required per stroke due to the pre-opening of the outlet valve. Below a certain speed, if the engine is run with wide-open throttle on the governor, a peculiar tendency to hunt is the result.

18 Among the means employed to prevent fluttering of lift valves the dash pot has always played a prominent part. Fig. 11 shows a typical outlet valve in connection with a dash pot or cushioning chamber. Regulating valves or pet cocks are placed above



1 in. represents 20 lb. sq. in. on original card
Dia. of cylinder 96 in.
Stroke of engine 66 in.
32 r.p.m.

FIG. 10 INDICATOR CARD TAKEN FROM ENGINE SHOWN IN FIG. 9

and below the cushioning piston to provide for any desired or necessary degree of sluggishness. This type of valve has been used by several engine builders.

19 Fig. 12 shows a design employing the outlet valve described in Par. 18. As usual the inlet valve is mechanically operated. It is of the rocking valve type and is so arranged that clearance volume is kept down to a small value. The power required for operating the inlet valve is quite small, because the latter moves very little when it is unbalanced. When this type of valve was first used by the present builders about ten years ago, the author expressed the opinion that the valves would be very short lived, owing to wear by the dust laden atmosphere of furnace plants. Actual experience, however, has proved that these valves do very well and that the wear is small. Free inlet areas amounting to $11\frac{1}{2}$ per cent of the piston area can easily be obtained by this valve and a greater per cent be secured by a slight increase in the clearance volume. The area through the

holes under the outlet valve disks is 16½ per cent of the piston area. To utilize this area to the fullest extent the outlet valves would have to lift 3½ in. For reasons which will be explained, a lift of this size is impractical, except for low rotative speeds, when it is of course not needed, so that in practice a lift of only 2¼ in. is used, cutting the available outlet area down to 11 per cent of the piston area. A large number of blowing engines with this valve gear are in successful operation both in the United States and in Canada.

20 What may be termed the most original of all blowing-engine valve gears is shown in Fig. 13a. Inlet and outlet valves are of the gridiron type of 1¾ in. stroke. The inlet is controlled by a cam

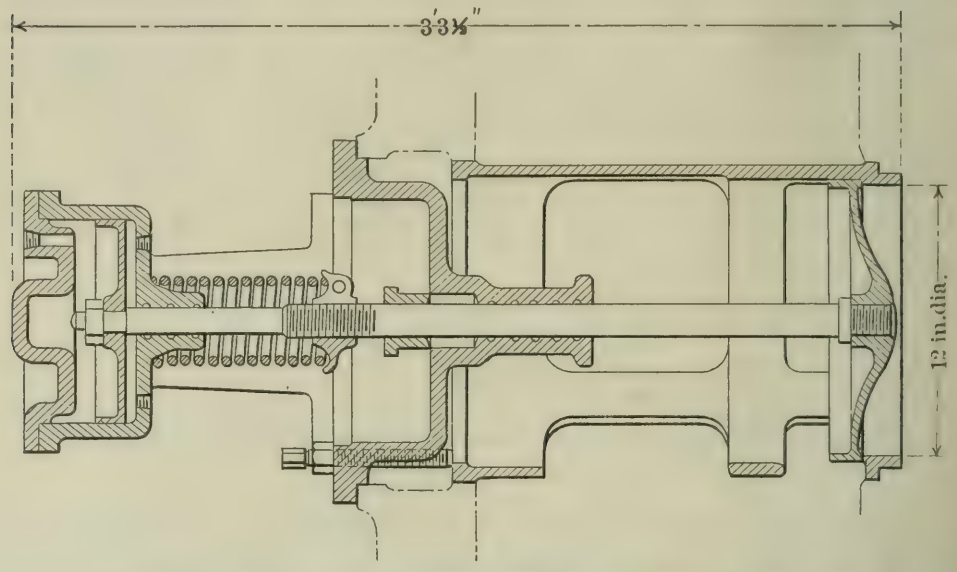


FIG. 11 AUTOMATIC DISCHARGE VALVE WITH DASH POT

acting in pair closure and the outlet valve is pulled open by the piston of an actuating cylinder. One side of the piston is open to atmosphere, the other side to the blowing cylinder. As compression proceeds in the latter, the friction holding the valve in position becomes less, and the force tending to displace it grows. The actuating cylinder is so proportioned that for a given speed of engine, for a given blast pressure and for a given method of lubrication, the valve opens at the correct instant. The kinetic energy of the moving valve is dissipated in a cam-operated dash pot which closes the valve at the dead center of the engine.

21 By this design fluttering of valves is most successfully avoided, large areas for inlet (14 per cent of piston area) and for outlet (11 per

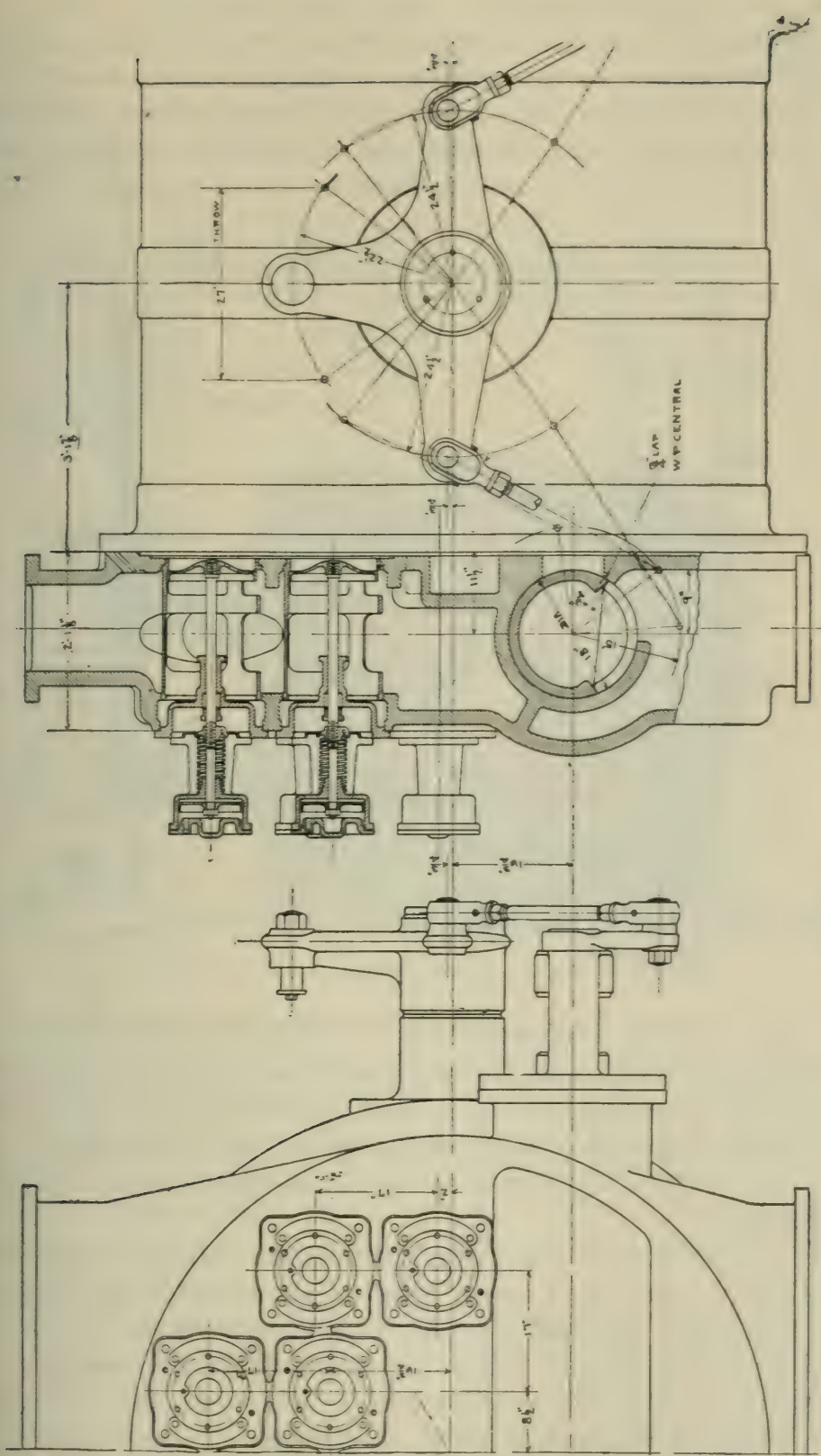


Fig. 12 Original Mesta Valve Gear

cent) can be obtained, and the power requirements are small. Ten and twelve years ago this type of valve gear was very popular in the United States and several engines of this type have been built in England and Belgium, but comparatively few have been constructed in recent years. The air cylinder needs large relief valves to prevent injury to the engine in case of accident to the valve gear. Owing to

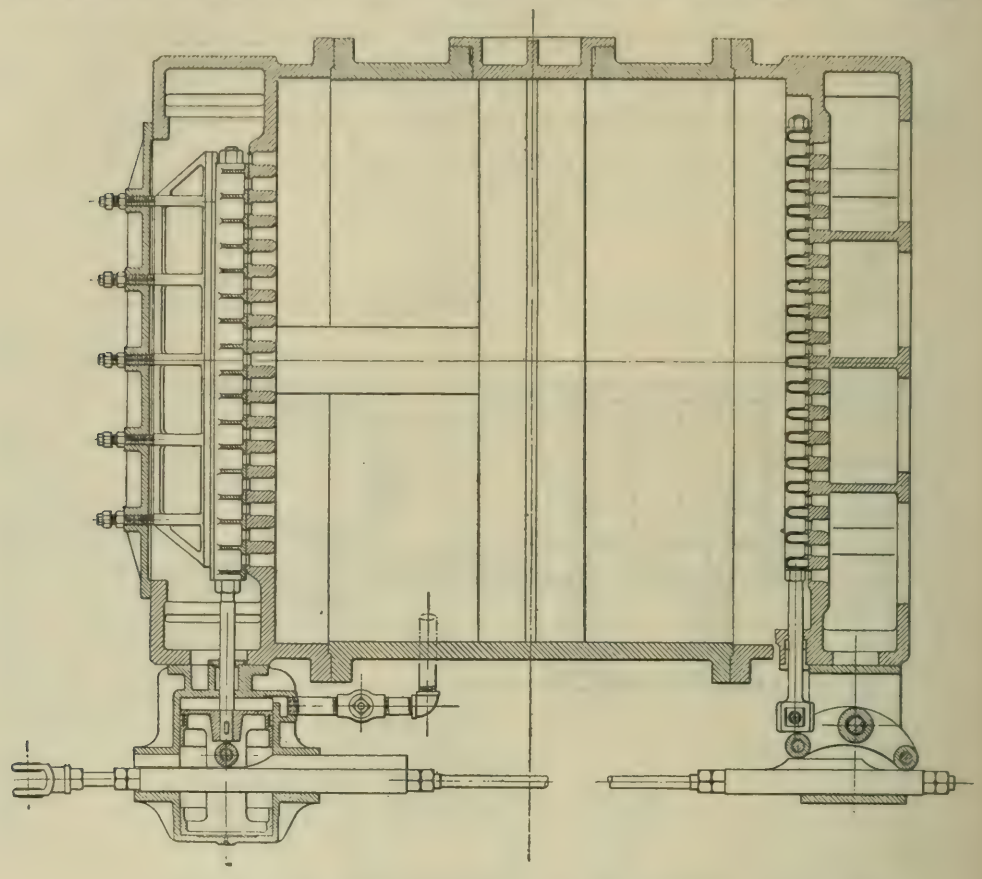


FIG. 13a SECTION THROUGH THE SOUTHWARK VALVE GEAR

the fact that at slow speeds the outlet valve opens much too early, more work is done per stroke at low speeds than at high speeds, even more than that done in the design of Fig. 9, and serious hunting occurs, when the engine is run on the governor with throttle wide open.

22 Every one of the valve gears so far described gives most satisfactory results at a piston speed of about 300 ft. per min. In Par. 2 the fact has been mentioned that the insistent demand for lower first costs requires higher piston speeds, not only temporarily in emergency cases, but continuously. Naturally, the experiment of running the standard types of valve gears at higher speeds was tried. Comparatively little trouble was experienced with the mechan-

ically operated inlet valves, except that in some of the designs the throttling loss is much greater than might be expected from Fig. 5. This is due to the passing of the air around corners in the valves, while it is flowing at high speed (see Fig. 6 and Fig. 7). Since it is easy to observe the inlet throttling loss on an indicator card by drawing the atmospheric line, it attracted attention and was probably given undue prominence. Phrases such as "small volumetric efficiency," and "inability to fill the cylinder" were often heard. The much greater, but invisible loss of volumetric efficiency, due to heat interchange between air and cylinder walls, was hardly ever mentioned in this connection.

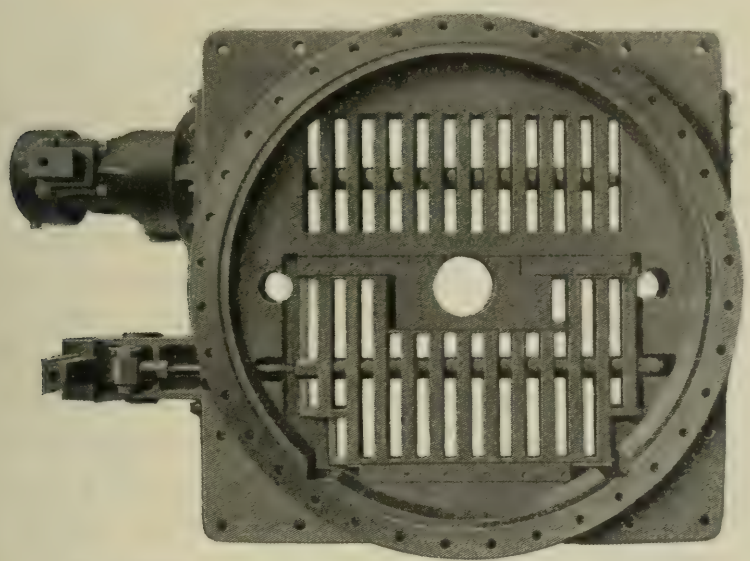


FIG. 13b VIEW OF SOUTHWARK CYLINDER HEAD

23 Thus the "standard" valve gears gave at 600 ft. per min. throttling losses ranging from 0.4 to 1 lb. per sq. in., and engineers were trying to increase inlet valve areas up to 20 per cent or more. At the discharge end serious troubles occurred with increase of speed. In the designs of Fig. 6 and Fig. 7 rapid wear of the valve seat and hammering of the valves occur. Whether this is due to a throwing of the valve against its seat by the plunger or to other causes is not certain.

24 In the design shown in Fig. 12, quietness of operation can always be enforced, even at the highest speeds, but the spring loading on the valve must be increased and its lift in the dash pot must be reduced. Considerable throttling loss is then encountered. All designs using large, heavy, high-lift valves experience in common an excessive hump on the card at the time of the opening of

the discharge valve, because it takes too long to provide the necessary area for discharge. The type shown in Fig. 13a and 13b is free from this evil, provided very large actuating cylinders are used, but their size only accentuates the trouble in regulation already mentioned.

25 If the American standard valve gears are used for 600 ft. per min. piston speed or above, inlet throttling losses of 3 to 6 per cent of the ideal blowing work occur, and outlet throttling losses of 7 to 12 per cent of the ideal blowing work occur. Besides, power for mechanical operation of valves increases, and other troubles of wear, breakage, or regulation appear, depending upon the valve gear.

26 One of the first men to judge the situation correctly was E. E. Slick of Pittsburgh. He realized that large areas are necessary and that the periphery of the cylinder may be used for obtaining area, if the cylinder head does not suffice. Attempts to use the periphery are old, but all of them introduced additional clearance volume. The design of Mr. Slick is free from this fault. Figs. 14 and 15 illustrate Mr. Slick's engine as built by two different firms. The cylinder proper or tub is made movable and serves as a mechanically operated inlet valve. Unobstructed inlet areas of 18 to 20 per cent are easily obtainable and practically without clearance space. The outlet valves in Fig. 14 are a modified form of the type shown in Fig. 6 except that the number of valves has been doubled. The outlet valves shown in Fig. 15a are also an invention of Mr. Slick. They are flexible plates and open against a curved guard. Here, too, large valve areas can be obtained and what is very important, they can be obtained with small valve lift.

27 For piston speeds up to 600 ft. per min. and for rotative speeds up to 65 r.p.m., the Slick tub has been very successful. The design has been severely criticized as "wagging the dog and holding the tail still" and the author confesses that he felt the same way when he saw the first Slick compressor more than ten years ago at the Edgar Thomson Steel Works, but the ingenuity of the design is forcibly impressed upon anybody who attempts to produce the same combination of large areas and small clearance space in some other way. If 65 r.p.m. are exceeded with this type, trouble begins. The inertia forces of the heavy cylinder are hard to take care of and heat the eccentric which moves the cylinder.

28 In the design of Fig. 14, wearing of valves and seats and hammering of the valves occur. In the design of Fig. 15, the valves break. They are not protected against lateral wind currents which exert a tearing action; furthermore the valves close late, and there-

fore with a slap, because no closing spring is provided beyond the elasticity of the valve plate itself.[1]

29 Another engineer who realized that the standard valve gears will not do for piston speeds of 600 ft. per min. is George Mesta of Pittsburgh. Fig. 16 illustrates the manner in which he obtained large

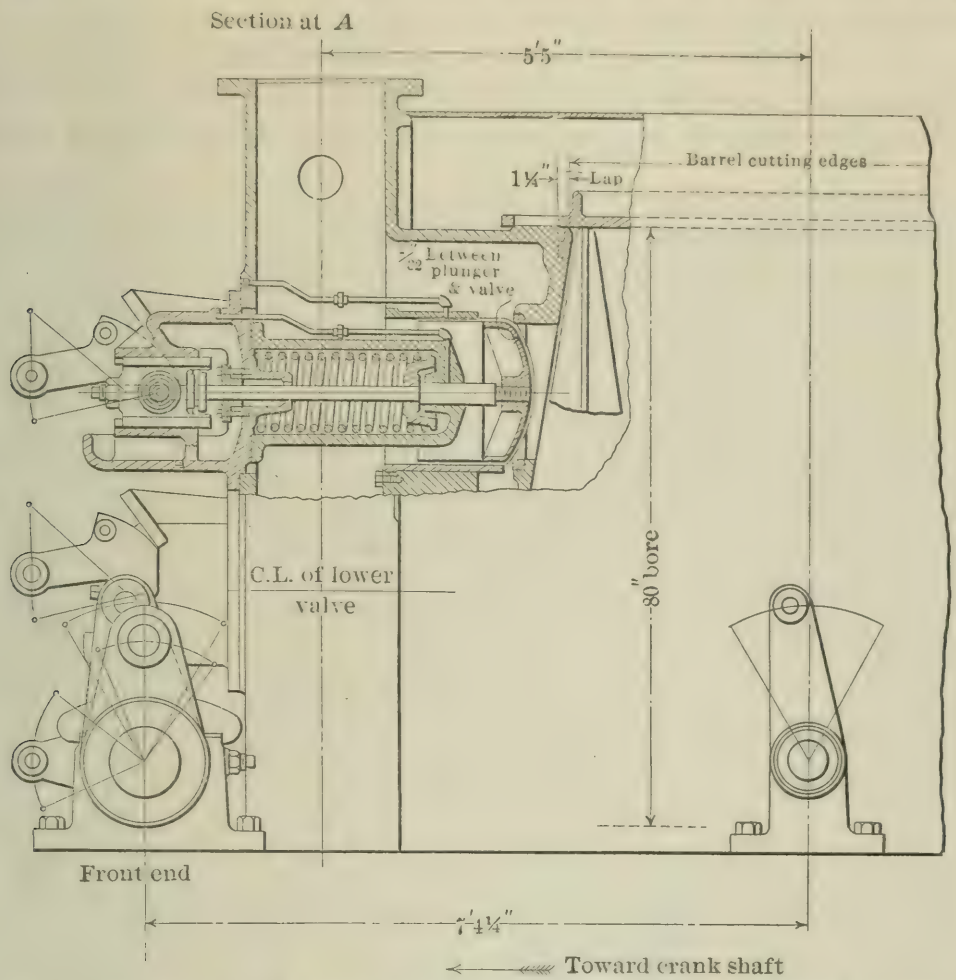


FIG. 14 BLOWING TUB WITH MOVABLE CYLINDER (SLICK DESIGN) BUILT BY ALLIS COMPANY

inlet and outlet areas and enforced quiet operation of valves. Rocking valves, two for each head, control both inlet and outlet; the inlet passes at the side of each valve, the outlet through the center of the valve. Automatic cup outlet valves are located beyond the rocking valves and are protected against the return closing slam by the mechanical closing of the rocking valves. This latter design has

[1] The engineers of the Snow Steam Pump Company, who also build blowing engines on the basis of Slick patents, advise the author that their firm makes the valve seat of Fig. 15b slightly curved, so that the plane valve rests against the seat with an initial tension. The result is that higher speeds can be attained without breaking of valves.

been used on vacuum pumps and compressors for over 20 years. Its adaptation to high-speed blowing-engine practice required doubling the valve equipment for the purpose of obtaining large areas without excessive diameter of rocking valve. The pot outlet valve is cushioned very little and is loaded lightly so as to fly out of the road of the blast without fluttering.

30 In Fig. 17 two indicator cards taken from an engine of this type are reproduced. It will be noticed that in spite of the large inlet valve area, 18 per cent, some throttling appears at 70 r.p.m.,

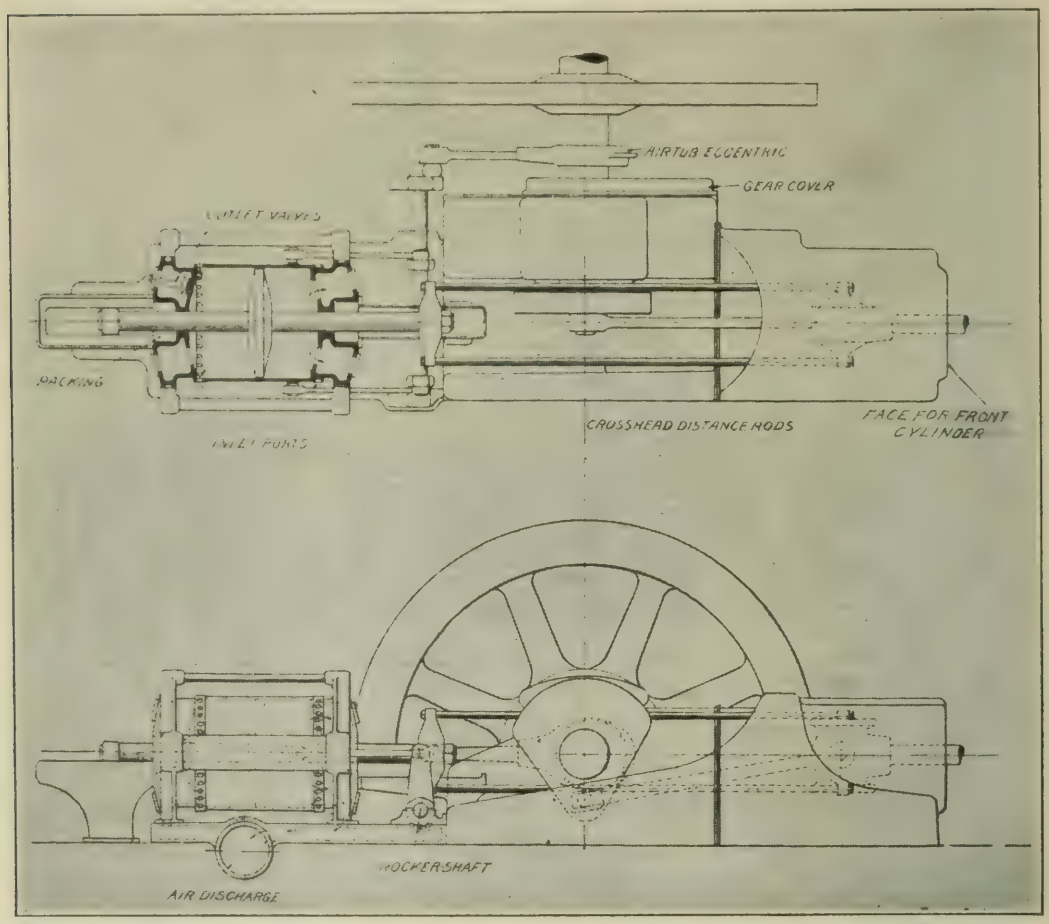


FIG. 15a BLOWING TUB WITH MOVABLE CYLINDER (SLICK DESIGN) BUILT BY WESTINGHOUSE MACHINE COMPANY

700 ft. per min. piston speed. The pressure line in the blast-receiver space has been entered, so that an idea may be gained of the outlet-throttling loss. Actually the loss is slightly greater than here shown, because calibration of the indicators revealed the fact that their scales were not quite alike.

31 The design of Fig. 16 has been very successful and has been used up to 820 ft. per min. piston speed, 82 r.p.m. with 60 in. stroke.

With the exercise of care in finishing the large rocking valves and their seats the resistance of these valves consumes less than 2 per cent of the ideal blowing work (air card). The principal drawback of this type of engine probably lies in its cost and in the oil consumption of the rocking valves.

32 In Europe the high-speed blowing engine is an accomplished fact. There the problem has been attacked along altogether different lines. European engineers long since realized that the harmful kinetic energy stored up in a valve is proportional to its mass and to its travel, and that, therefore, both should be cut down. This

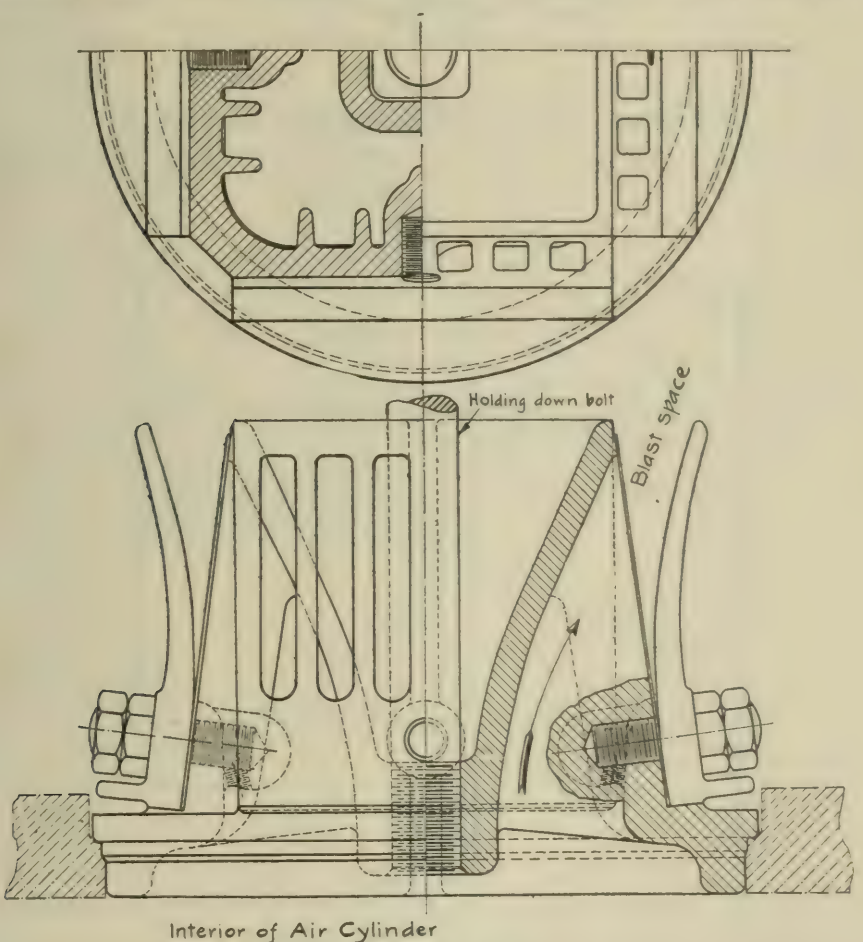


FIG. 15b SLICK AIR VALVE

fact in itself is not new to American engineers. The Weimer blowing engine (Fig. 18) which can look back upon a respectable number of decades, has tried hard to embody the principle of small mass

[1] Since this paper was written the author has been informed by Mr. Weimer that his company has been using low-lift aluminum valves for the outlet for about ten years and that they are quite successful for speeds up to 60 r.p.m. However, the guiding of the valves is not free from uncertain friction. This fact undoubtedly limits the rotative speed of the engine.

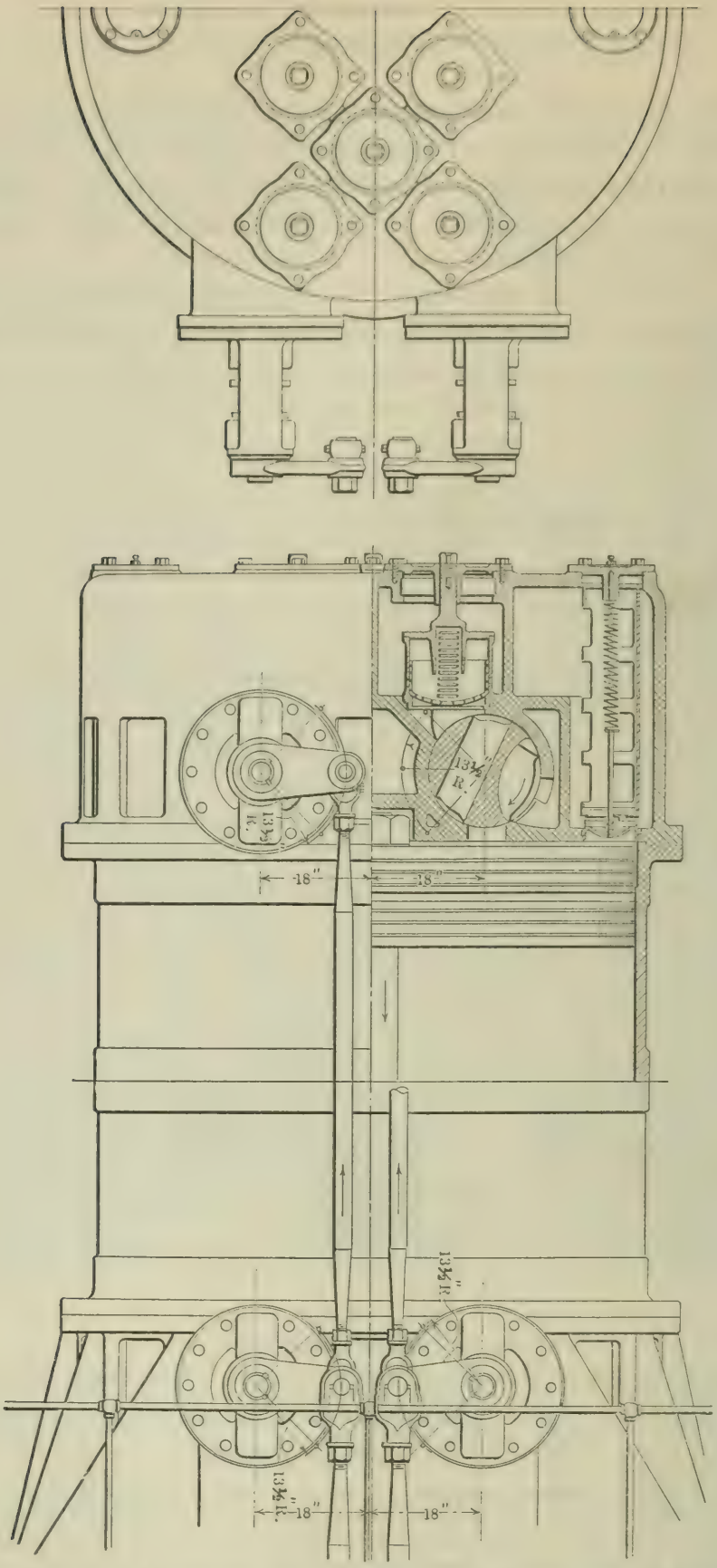


Fig. 16 Blowing Cylinder with Combination Inlet and Outlet Valve (Mesta)

and low lift. It has failed by sticking to the use of leather which is unsuited to high pressures[1]. Fifteen years ago Mr. Mansfield designed a blowing engine for Leetonia, O., in which he used a great number of light, low-lift strips. The engine probably would have been a success if it had not been so weak for the blast pressure. Mr. Slick (see Fig. 15) introduced light low-lift valves. But these attempts cannot compare with the sweeping success on the other side of the Atlantic.

33 Out of the mass of European designs, two stand out conspicuously, the Hoerbiger (Fig. 19) and the Borsig (Fig. 20). As will be noticed, these valves are of the automatic, multi-ported, low-lift type. They are guided without friction by elastic deformation of part of the valve. To get enough valve area, the cylinder head must be extended considerably beyond the piston diameter (Fig. 21), or a valve belt must be used and the clearance corre-

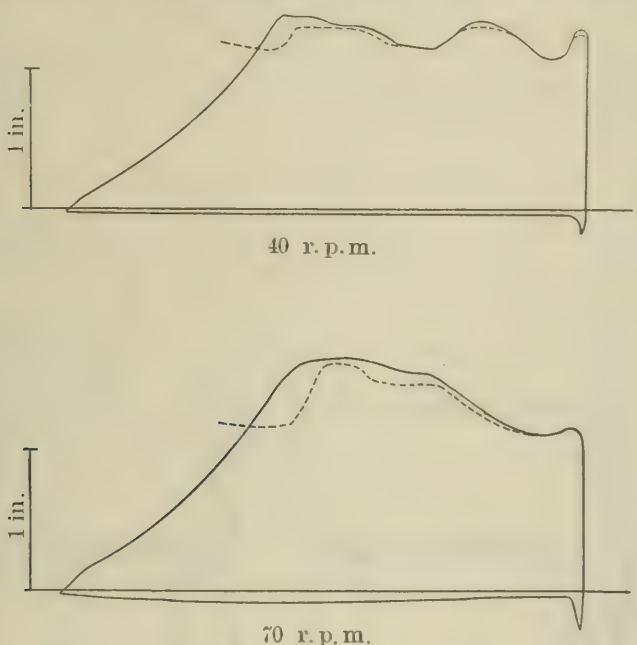


FIG. 17 INDICATOR CARDS TAKEN FROM ENGINE SHOWN IN FIG. 16

spondingly increased. It should be noted that there is no novelty in the valve belt as such. It was used 20 years ago, both in this country and abroad, and was temporarily abandoned because no satisfactory automatic valve existed.

34 European engineers do not hesitate to use large clearance spaces if by doing so other advantages can be gained, and they meet with success. Matters are different in this country. Clearance in a blowing engine seems to be an eyesore to the American furnace

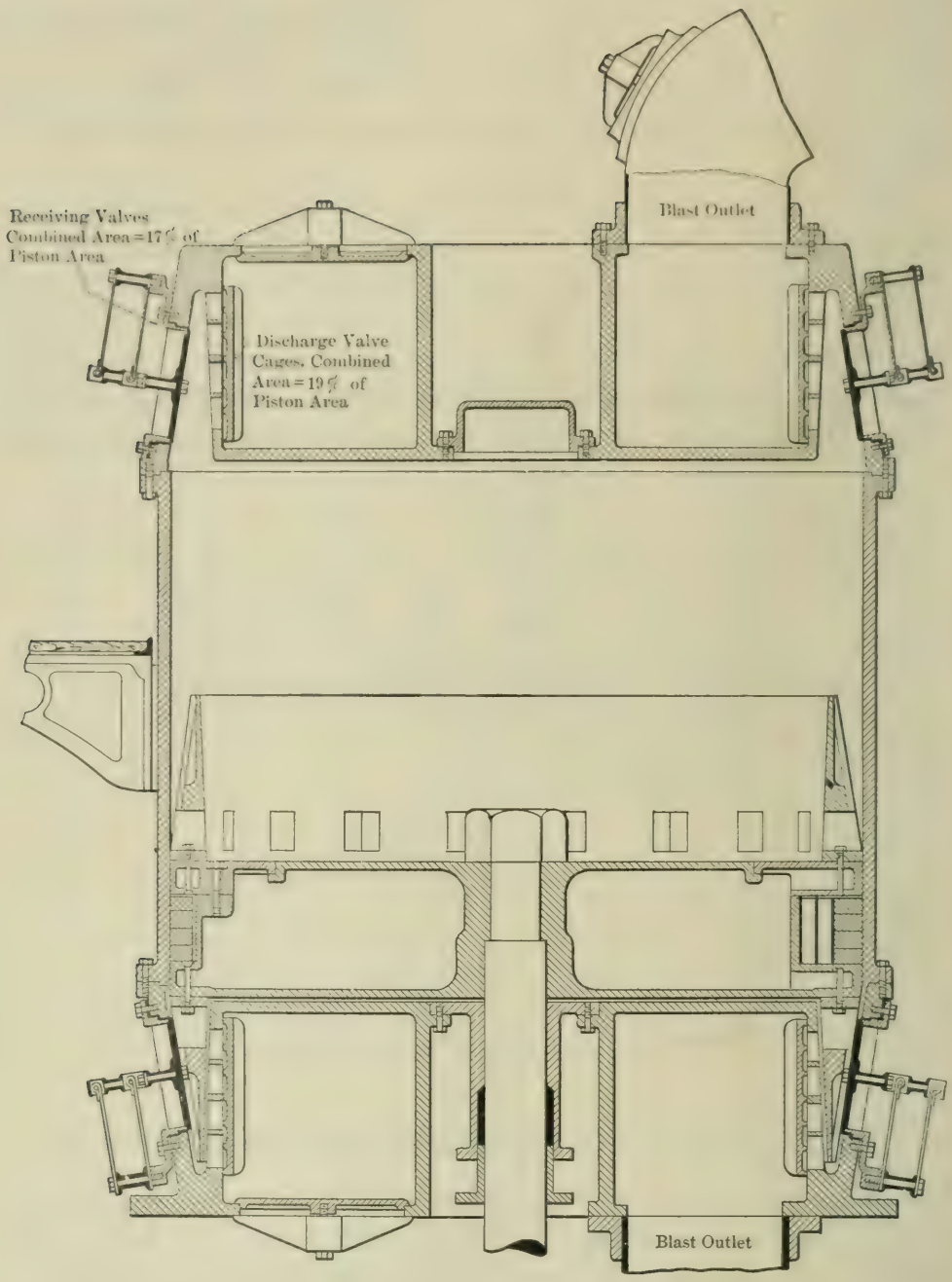


FIG. 18 ENGINE WITH AUTOMATIC INLET AND OUTLET VALVES (WEIMER)

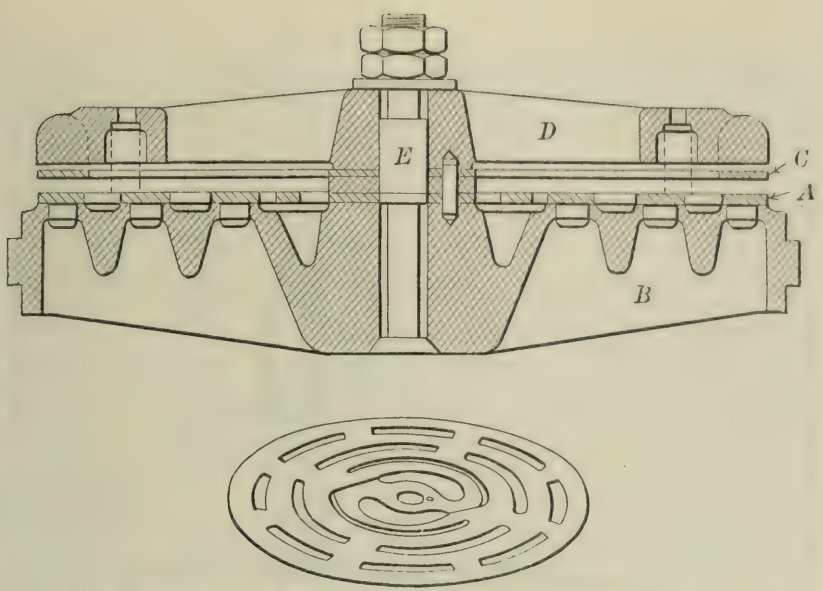


FIG. 19 MULTI-PORTED PLATE VALVE (HOERBIGER-ROGLER)

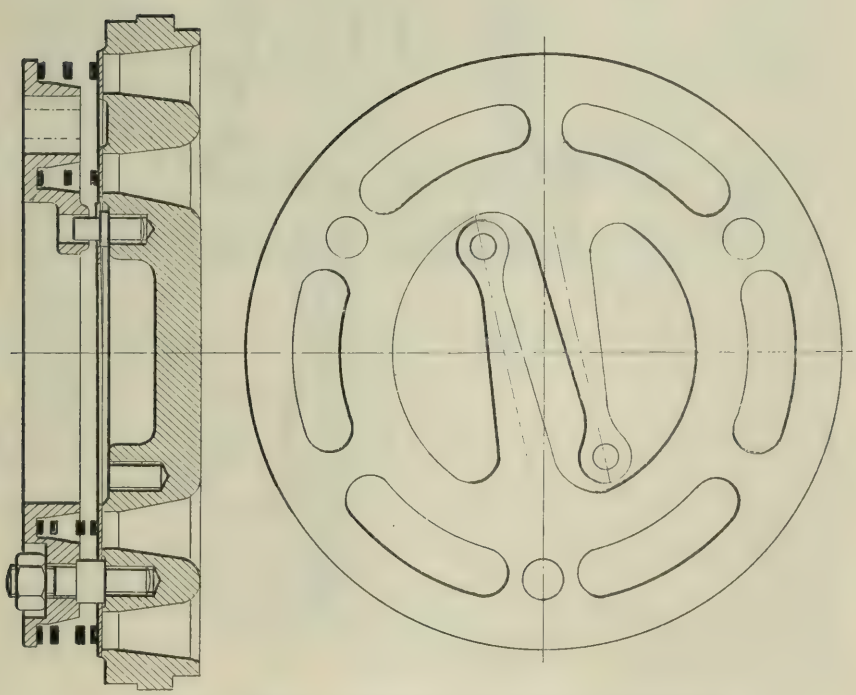


FIG. 20 MULTI-PORTED PLATE VALVE (BORSIG)

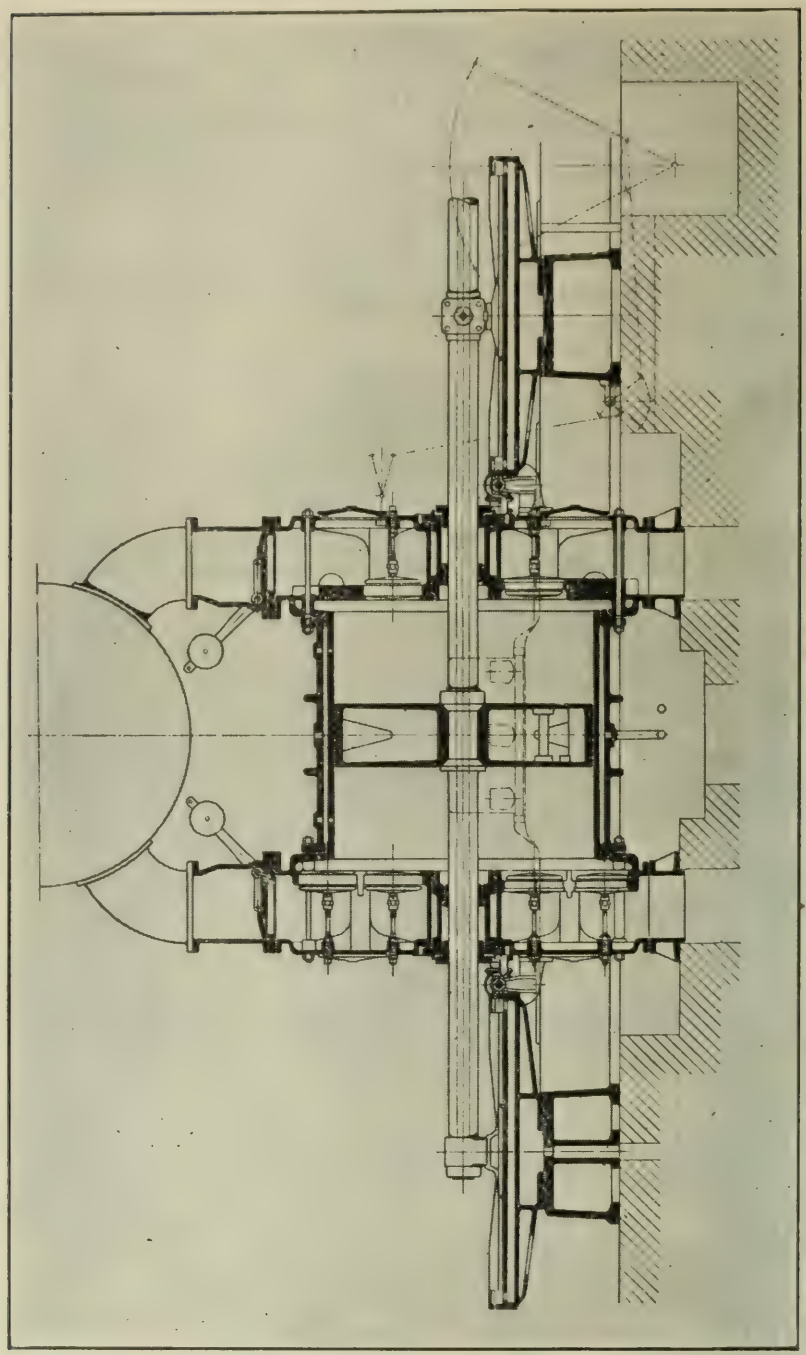


FIG. 21 BLOWER WITH MULTI-PORTED PLATE VALVES, SHOWING EXTENDED HEAD

man. The influence of this much abused clearance space can be summed up in a few words:

a Clearance volume increases the necessary size of blowing tub for a given weight of air to be pumped per stroke.

b The larger size of blowing tub results in a small increase of friction work and, therefore, in a larger size of power cylinder.

c The influence of the increased heat-exchanging surface on the true volumetric efficiency is small.

35 On the other hand, clearance allows the use of very large valve areas, which fact decreases throttling work and causes better filling of the air cylinder and also allows higher piston speeds, or in other words, a smaller and cheaper engine. The higher piston speed makes possible the use of a more efficient prime mover, namely, the gas engine. When the truth of this is realized, recognition of the merits of the modern European high-speed blower should present no difficulties. The plate valves are so light in weight and the spring load can be made so small that for the greater part of their working time the valves rest against the guard or stop; this fact, of course, greatly reduces fluttering. Furthermore, there are no wearing parts and, therefore, no sliding surfaces and neither sticking nor binding from gummed and dusty oil. The low lift does not allow the valve to acquire destructive velocity in closing. If a sufficient number of valves are used the pressure loss through the valves is small and the filling of the cylinder is most perfect. The life of the valves is long, provided that they are made of the proper high-grade steel and that the spring loading is properly proportioned. If a valve should break, it can easily be replaced because the valves are light; besides the inlet and outlet valves are alike, so that only a few valves need be carried in stock.

36 To engineers who are accustomed to standard American practice some of the foregoing statements will appear too good to be true and will require proof. Fig. 22 shows an indicator card taken from a Borsig blowing engine in Differdingen. Fig. 23 shows two cards taken at 55 and 82 r.p.m. from a blower with Hoerbiger-Rogler valves in Rombach. This latter card shows the blast pressure in the main just outside the valves, and allows the computation of the valve losses. In Fig. 24 these losses have been plotted in per cent of the ideal blowing-engine work against piston speed. In the same illustration are shown the valve losses from the Differdingen-Borsig blower computed from detail drawings, and for

comparison, the valve losses of a Mesta high-speed blower using cup valves, the results of actual test, are given. The calculation of the Borsig valve losses was based upon the test of the Mesta engine, that is to say, the method of figuring valve lifts and losses was checked by a test and the coefficient of discharge was thereby determined. The discrepancy between inlet and outlet losses on the Hoerbiger-

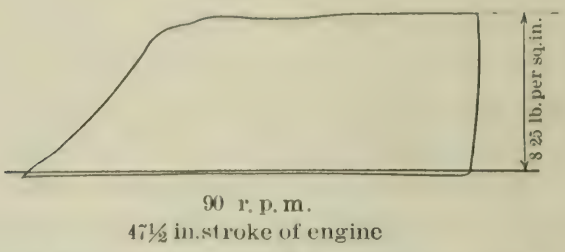


FIG. 22 INDICATOR CARD TAKEN FROM CYLINDER WITH VALVES SHOWN IN FIG. 20

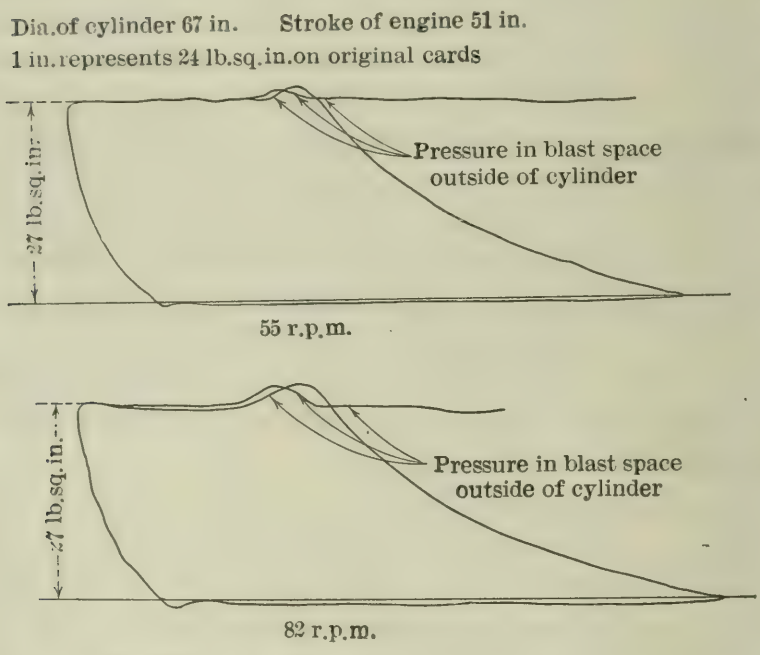

Dia. of cylinder 67 in. Stroke of engine 51 in.
1 in. represents 24 lb. sq. in. on original cards
27 lb. sq. in.
Pressure in blast space outside of cylinder
55 r.p.m.
27 lb. sq. in.
Pressure in blast space outside of cylinder
82 r.p.m.

FIG. 23 INDICATOR CARD TAKEN FROM CYLINDER WITH VALVES SHOWN IN FIG. 19

Rogler and Borsig valves is entirely due to the respective number of valves employed. It will be observed that the plate valves show a very much smaller loss than the cup valves, although the cup valve area is very much larger than standard American practice.

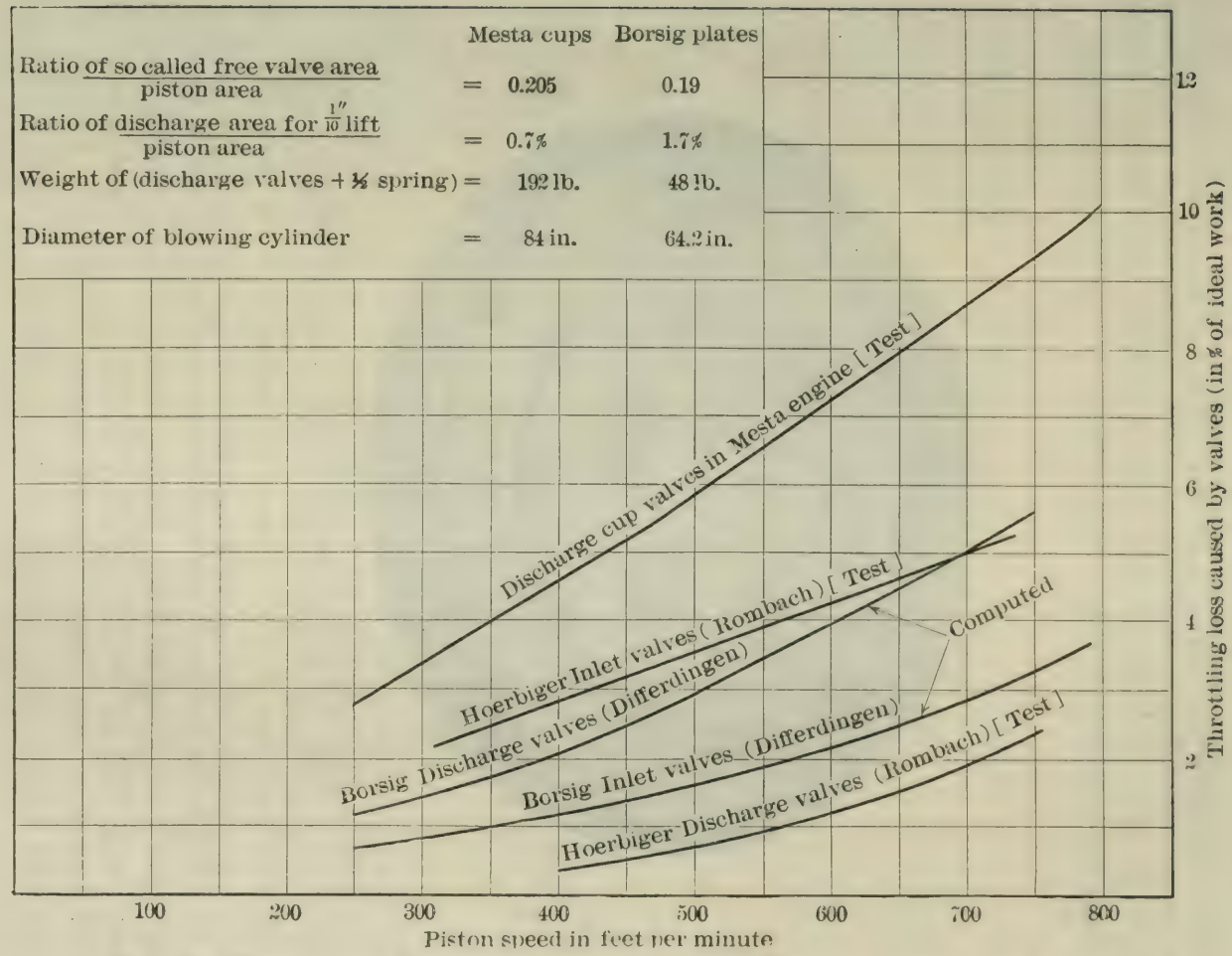


FIG. 24 THROTTLING LOSSES CAUSED BY VARIOUS TYPES OF DISCHARGE VALVES

Comparison is invited between the engine (Fig. 16) whose discharge valve losses are represented by the chart, and between the engine shown in Fig. 14. The former engine has an 84-in. cylinder bore and ten cup valves of 12-in. diameter. The latter engine has an 80-in. cylinder bore and four cup valves of 18-in. diameter. Since the area exposed per unit lift, say 1-in. lift, counts in a discharge valve, it will be noticed that our American high-speed blowing engines are apparently somewhat inferior to the European type.

FIG. 25 MULTI-PORTED PLATE VALVES (MESTA)

37 With regard to the life of these valves, statements of European builders are interesting. Borsig asserts that in several engines there has not been a single broken valve in spite of continuous high-speed operation, and that in places where valves have broken, this fact could invariably be traced to bad material. Haniel and Lueg, builders of blowers with Hoerbiger-Rogler valves, make exactly the same statement. Particular emphasis is placed upon the almost silent operation of these valves, both by users and builders. No separate cushioning means are employed except that in the Hoerbiger-Rogler valve an elastic plate softens the impact of the opening stroke before the valve strikes the guard. This cushioning alone does not suffice, but another circumstance comes in helpfully.

Thin films of oil coat the valve plate, cushion plate and guard. The squeezing of the air and oil between these plates provides a sufficient cushion to prevent injury to the valve.

38 The author has taken particular pains to secure data on the behavior of valves. Unfortunately, it was impossible to secure such data on the Borsig and Hoerbiger valves with the exception

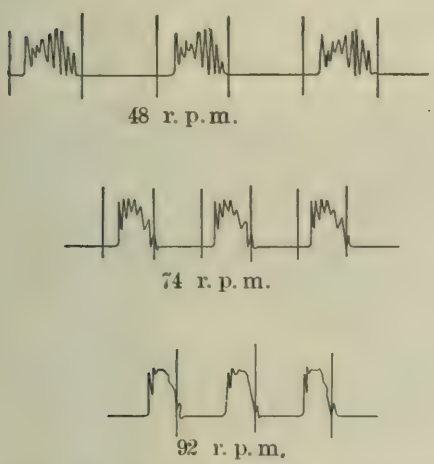


FIG. 26 VALVE MOTION DIAGRAMS TAKEN FROM INLET VALVE SHOWN IN FIG. 25

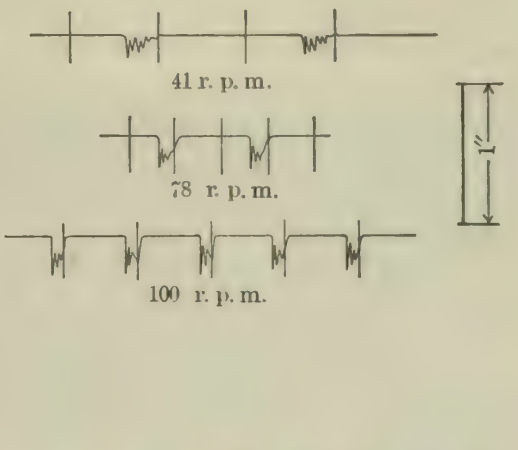


FIG. 27 VALVE MOTION DIAGRAM TAKEN FROM DISCHARGE VALVE SHOWN IN FIG. 25

of the indicator cards, Figs. 22 and 23, which, however, are only circumstantial evidence. The author considers himself fortunate in having secured valve behavior diagrams from an experimental engine of the Mesta Machine Company, which has lately begun a series of systematic experiments on a plate valve patterned after those of Hoerbiger and Borsig. This valve, Fig. 25, designed by L. Iversen, is intended to equal, or if possible, even surpass the European valves. It is guided without friction by a volute spring, flies up against a cushioned guard and is in all respects similar to the above described European valves. Valve motion curves taken from this valve may, therefore, be regarded as indicative of the whole type.

39 In Fig. 26 the diagrams of the inlet valves are given, and in Fig. 27 the diagrams of the discharge valve. A standard blowing-engine valve was used, which meant in this case that the valve area was larger than necessary. The result is that the inlet valve flutters somewhat at low rotative speeds and that the discharge valve lifts very little. Besides, the spring load was too great. Allowing for

these departures from correct conditions, the curves present enough interesting features. Turning to Fig. 26, it will be noticed that the time of vibration of the valve is practically constant, so that at the high speed the valve has not time to perform a great number of these vibrations per engine stroke; in fact, only one small vibration during lifting and one small vibration during closing occur. At most speeds the valve is closed on the dead center, only at the very highest speeds it rebounds and closes a trifle late. This rebound appears to be caused by a sort of compression wave under the valve, or rarefaction wave above the valve so that the rebound is not necessarily caused by elastic compression of the valve seat. The velocity with which the valve strikes the seat can be determined approximately from the diagrams by the tangent method.

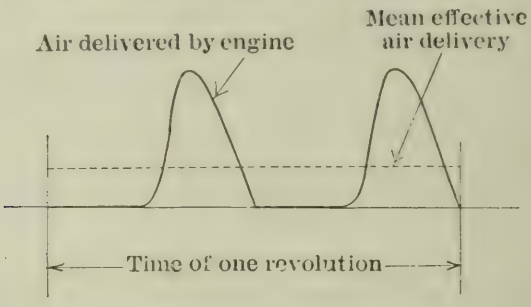


FIG. 28 AIR-WAVE PULSES CAUSED BY RECIPROCATING BLOWER

40 From the sound of closing of these valves, the author judged that their velocity of striking must be very much less than those of cup valves in the designs of Figs. 5 and 6. For this reason he was surprised to find from the diagrams (Fig. 26) that the velocity is quite high enough to call for very good material, such as alloy steel. This would be exactly in line with the experience of the German builders who state that unqualified success in these valves is not only a matter of design but just as much a matter of good material.

41 From Fig. 27 it will be observed that the outlet valve is reflected by the cushion plate and is thrown back into the air current, where it duly performs its vibration. At several speeds it closes late. The air cards, however, show no signs of slip. In spite of the slight delay, the outlet valves close very quietly, as is proved by sound and by the rounding of the diagram at the closing point. The vibrations of this type of valve are of so short a duration that the air in blast mains does not set up synchronous vibrations. The tests on this valve have not yet been finished.

42 From a study of the various types of valves and valve gears, it appears that at the present time the low-lift, alloy steel plate valve promises to become the standard valve for high-speed blowing engines, because (*a*) there is no wear or binding or sticking; (*b*) no lubrication is required; (*c*) it causes very small throttling losses; (*d*) it can be used for the highest speeds; (*e*) it is inexpensive; (*f*) it does away with mechanical gearing, oiling and adjustment.

43 No matter with what valves a reciprocating blower is equipped, its delivery remains discontinuous, that is, it delivers air impulses comparable to a constant delivery, over which is super-

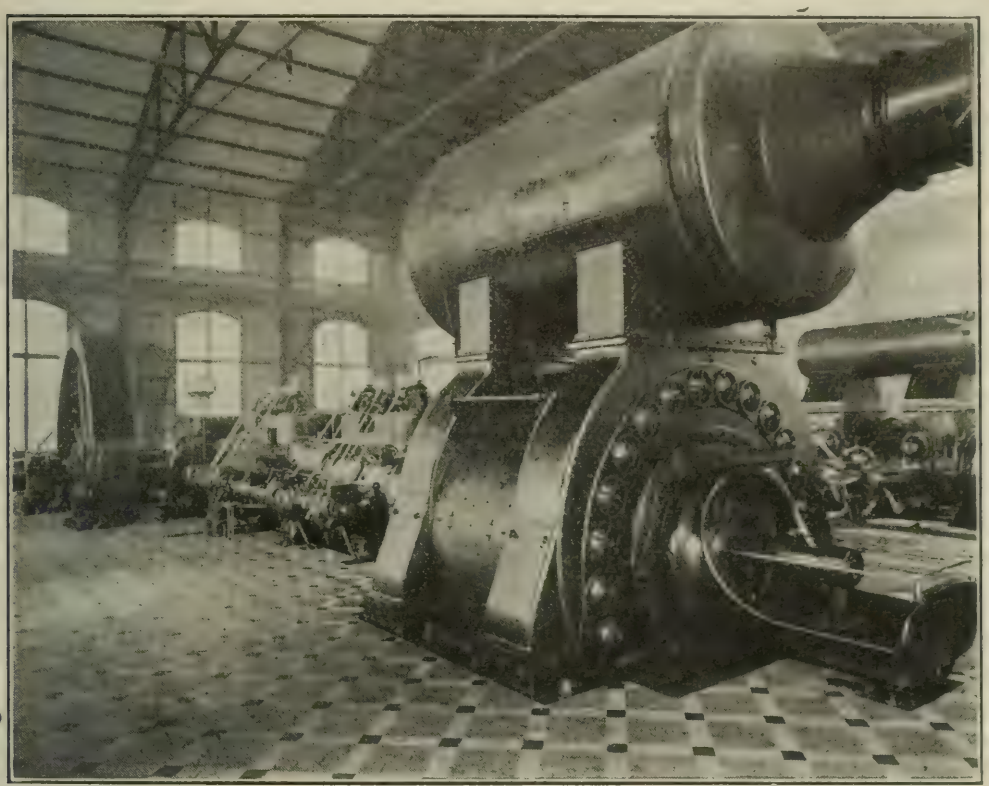


FIG. 29 BLOWING ENGINE WITH AIR TANK ON CYLINDER

posed a wave motion or vibration (Fig. 28). If the blower discharges directly into the blast main, then vibrations are transmitted with undiminished strength and shake the whole line. In steam-engine practice this evil was long ago cured by placing a large steam or water separator near the engine, and so convinced are we of the necessity of this separator that we install it also on lines carrying superheated steam, ostensibly for the purpose of dropping out moisture which is not there, but really to damp the vibrations of the

pipe line. If a similar request is made of a furnace man for the air line, a great deal of resistance is encountered. The author knows of only one furnace plant in this country where a large tank or equalizer was installed for each blowing engine. The pipe lines thus connected are practically free from vibration.

44 In Europe a similar resistance is probably offered by furnace men; at least it would appear so from the fact that modern high-speed reciprocating blowers are equipped by the builders with large tanks resting on the backs of the cylinders, thus permitting the furnace man to retain his long cherished ideas of pipe-line design (see Fig. 29). The results must be satisfactory, if comparison is made with the American plant in which tanks are used.

45 Summing up, we find that the reciprocating blower has in the last decade made wonderful strides towards becoming a successful high-speed machine. While the increase of piston speed was started by the gas engine as a matter of necessity, it has also benefited the steam-driven blowing engine, and isolated furnace plants can now work with two air cylinders instead of three, because one will successfully blow a furnace in case of emergency, or else three smaller units may be used.

46 The combination of the high-speed reciprocating blower with the blast-furnace gas engine makes the use of the latter profitable even in the Pittsburgh district where coal is cheap. The latest group of furnaces in this region has been equipped with slow-speed reciprocating steam-driven blowers. If a high-speed gas-driven blower had been on the market, the result would probably have been different, because then the first cost of the more efficient gas engine would have been lower.

47 A gas-driven blowing engine with 800 to 900 ft. per min. piston speed and high rotative speed will be the most formidable competitor of the turbo-blower, if European experience may be taken as a guide. There are engineers in this country who have already carried into practice higher piston speed for gas engines for electric power, and interesting developments in this line of work may be expected in the next five years.

APPENDIX No. 1

THE INFLUENCE OF CLEARANCE (VOLUME AND SURFACE) IN BLOWING ENGINES

48 The influence of clearance is fourfold:

a The high-pressure air imprisoned in the clearance space re-expands and prevents the entering of atmospheric air until the pressure in the clearance space has dropped down to atmosphere.

b The surface of the clearance space is heated by the compression, gives up heat to the incoming atmospheric air and thereby reduces the weight of air taken in per stroke.

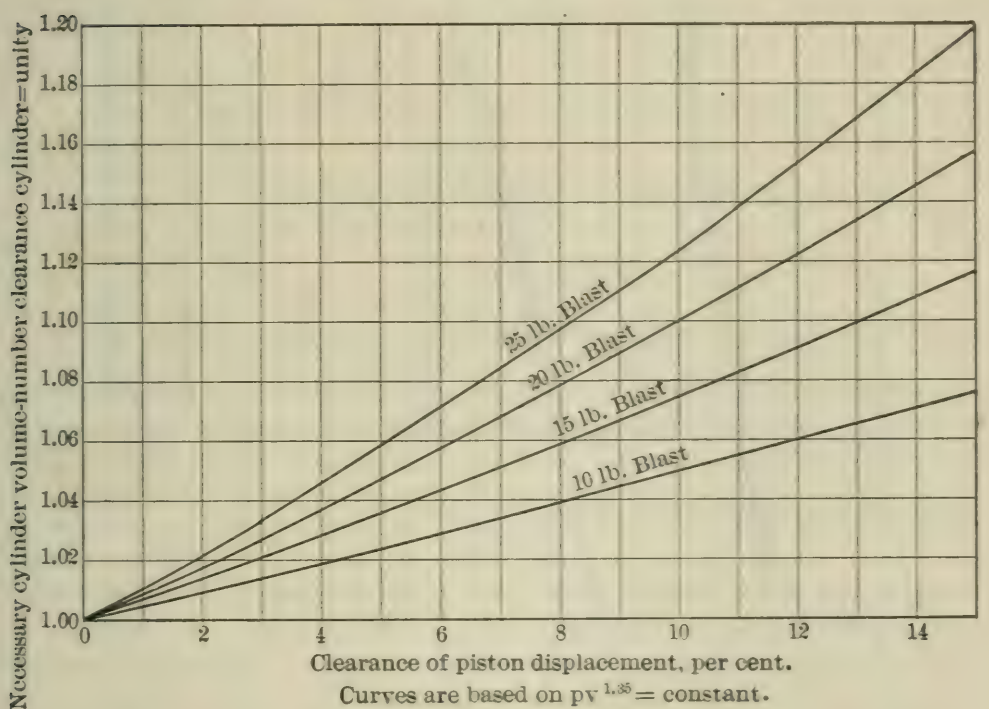


FIG. 30 INFLUENCE OF VOLUME OF CLEARANCE SPACE UPON PISTON DISPLACEMENT OF BLOWING ENGINES

c The two foregoing actions require a larger cylinder than would be required without clearance space or surface for delivery of a given quantity of air per stroke. The larger cylinder means heavier piston and rod, larger valves; briefly, a more expensive machine for unit weight of air taken in per stroke.

d The larger parts cause a loss by friction and this fact calls for an increase in the size of the power cylinder, steam or gas.

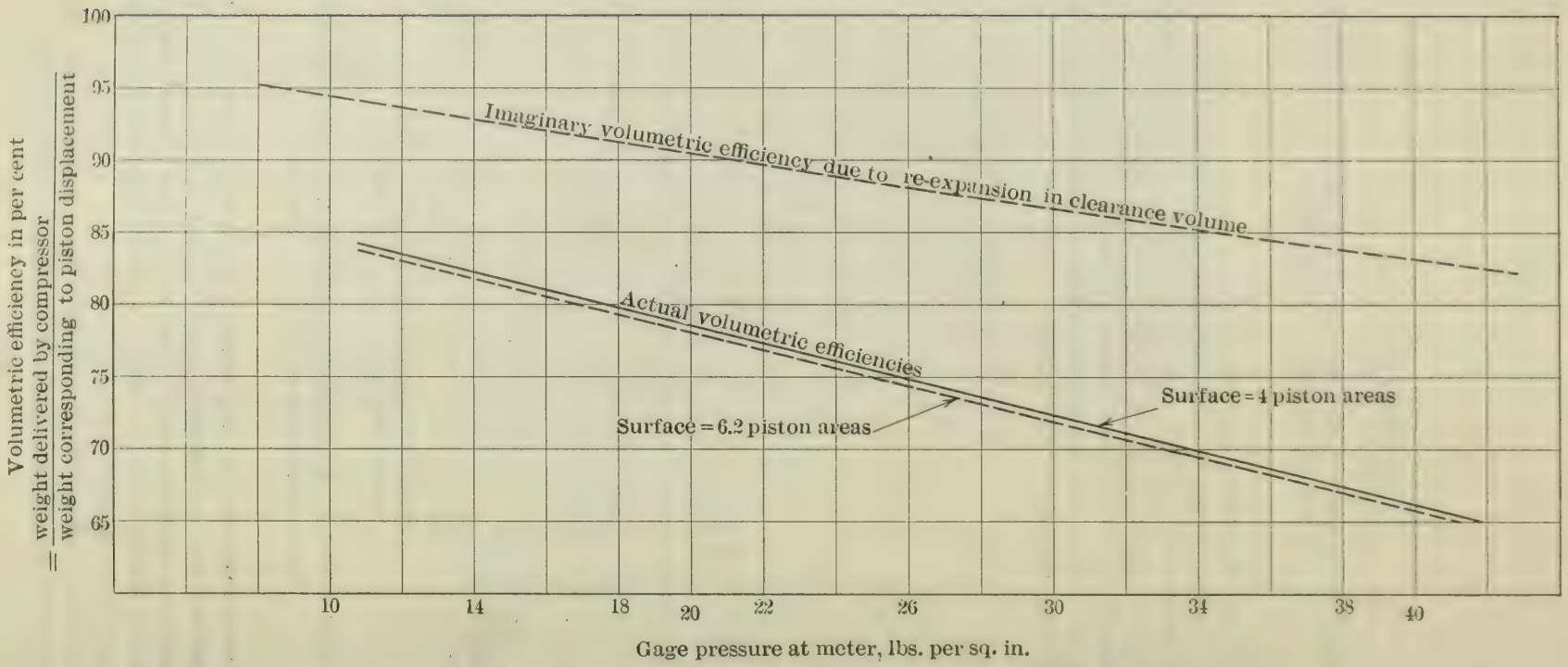


FIG. 31 INFLUENCE OF ADDITIONAL SURFACE IN CLEARANCE SPACE UPON VOLUMETRIC EFFICIENCY. CLEARANCE VOLUME =9.6% OF PISTON DISPLACEMENT

49 The enlarging of the air cylinder due to the action described under the first heading, re-expansion, can be mathematically determined. The results of such a calculation are plotted in Fig. 30. Since 15 lb. per sq. in. represents the average blast pressure, approximately 1 per cent of increase of cylinder diameter will be necessary for every 3 per cent increase of clearance volume.

50 The influence of clearance surface on weight entering per stroke cannot be mathematically treated, at least not by any formula simple enough for practical use. It should be noted that this harmful surface is present even if the clearance volume is reduced to zero. The lack of knowledge of the influence of the heating surface offers the most assailable point to the opponents of the reciprocating blower. Tests are therefore needed, and the author believes that on an engine with tight valves a thermometer in the intake pipe, a thermometer in the outlet immediately beyond the valves and an indicator will give a very good idea. From the temperature of the outlet and the shape of the indicator card, the temperature at the beginning of the compression can be computed. Comparison of the value thus obtained with the reading of the

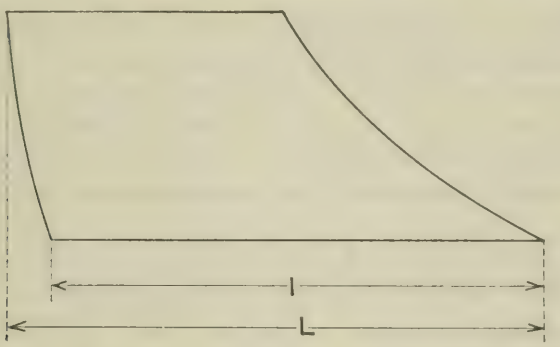


Fig. 32 Theoretical Indicator Card Showing Effect of Clearance

thermometer in the intake pipe gives the influence of the walls. From figures on the number of revolutions per pound of coke delivered to the blast furnace and from other data furnished the author by Julian Kennedy of Pittsburgh, it appears that the influence of these walls on large size engines is quite small indeed, but no definite conclusions can be drawn, because the reactions in the blast furnace vary.

51 The author was particularly interested in variations of volumetric efficiency caused by variations of surface of clearance space. To obtain data on this point, a small 8 in. by 8 in. compressor at the Carnegie Technical Schools was so equipped that its clearance surface could be varied, leaving the clearance volume constant. The quantity of air passing through the compressor was measured by a positive air and water tank displacement meter. Fig. 31 proves that the influence of extra surface is indeed very small, so small as to be of no practical influence in blowing-engine practice. This is reasonable since during the suction stroke the incoming air is exposed in a machine without clearance to heating by the cylinder head, the piston and approximately one-half of the cylinder barrel, and compared with this enormous surface the extra walls of ordinary clearance spaces are small. Besides, heat transmission decreases with diminishing temperature difference, and after the air has been heated a certain amount, further temperature rise becomes sluggish.

52 It should be noted that in this test conditions were purposely made unfavorable. The surface was large compared with the volume; the extra surface was arranged in the current of air; no radiation from these surfaces to the outside air could take place, and the compression was higher than usual in blowing engines. In view of these facts the statement that the influence of the walls of extra clearance space in blowing engines is negligible will appear justifiable.

53 A brief consideration of (*c*) shows that it is not so serious as might appear at first glance. The larger cylinder is imaginary, because if clearance is enlarged for the purpose of increasing the valve area with a view to allowing higher piston speed the actual cylinder volume per unit quantity of air pumped is much reduced.

54 If, for instance, the rotative speed of an engine is doubled, the ideal cylinder volume per unit weight of air delivered is cut in two; if the clearance is increased a few per cent to accommodate larger valves, the new unit cylinder volume will probably be 52 to 54 per cent of the original one, so that in spite of the theoretical increase of volume, a practical reduction, which is very material, results.

55 The friction loss mentioned under the fourth heading exists in reality and would doom clearance space, if no compensating benefit existed. The magnitude of the friction loss might be determined by tests, but in the absence of tests that cover a sufficient range, the following reasoning is suggested: The mechanical efficiency of blowing engines varies from 85 to 92 per cent, that is to say, the indicated air work is 85 to 92 per cent of the indicated steam work. In order to cover all contingencies, the calculation should be based upon the engine with greatest friction work, that is to say, an engine with 85 per cent mechanical efficiency; the latter figure is equivalent to the statement that 17.6 per cent of the indicated air work is friction loss. Let this figure be correct for an engine without clearance. Then the friction work in an engine with clearance will be greater principally for two reasons:

a If the stroke remains the same, the forces will be greater in the ratio of $\frac{L}{l}$ (see Fig. 32). Since the diameters of crank pin and of shaft in main bearing are nearly proportional to $\sqrt{\frac{L}{l}}$, the friction path is larger in ratio $\sqrt{\frac{L}{l}}$ for the machine with clearance. The work lost by friction will therefore be proportional to $\left\{\frac{L}{l}\right\}^{1.5}$.

b Piston area grows proportional to $\frac{L}{l}$, and the piston thickness grows proportional to $\sqrt{\frac{L}{l}}$, so that the piston weight which produces friction grows with $\left\{\frac{L}{l}\right\}^{1.5}$. These figures are based on equal strokes for machines with and without clearance. Friction work from this source, therefore, is also proportional to $\left\{\frac{L}{l}\right\}^{1.5}$.

56 Other friction losses, for instance those caused by crosshead sliding, flywheel windage and others, are not proportional to $\left\{\frac{L}{l}\right\}^{1.5}$, but they certainly do not exceed that ratio.

57 As a first approximation the friction loss may, therefore, be taken as proportional to $\left\{\frac{L}{l}\right\}^{1.5}$ (see Fig. 32). On the basis of a mechanical efficiency of 85 per cent for the no-clearance machine, the additional friction work due to clearance volume has been plotted in Fig. 33. It will immediately be recognized that an increase of clearance volume from 2 per cent to 6 or 8 per cent means a very small loss, a loss that need not be considered, if other more important features can be gained. The most important feature is a smaller and

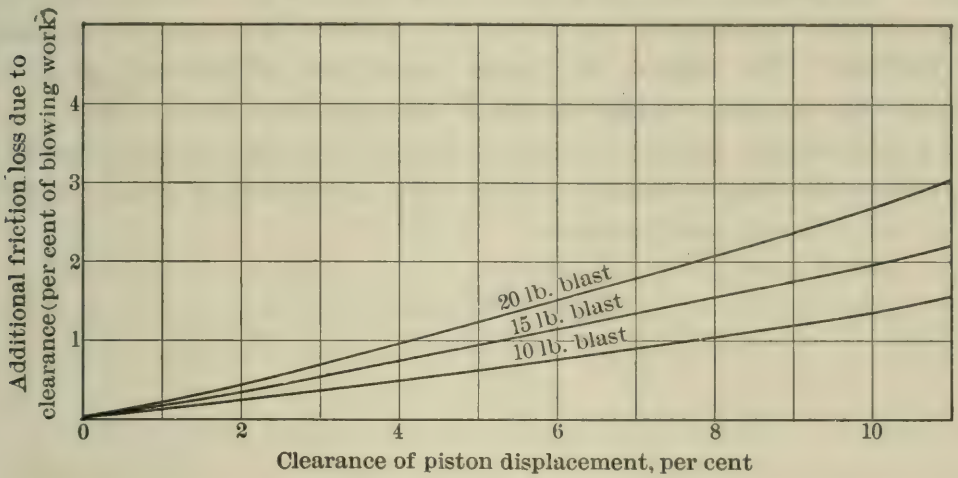


FIG. 33 ADDITIONAL FRICTION LOSS DUE TO CLEARANCE IN BLOWING ENGINES. BASED ON 85 PER CENT MECHANICAL EFFICIENCY OF ENGINE WITHOUT CLEARANCE

cheaper engine. In steam-driven units this in itself is a gain from the standpoint of the purchaser, provided that 3 or 4 per cent in fuel consumption makes no difference. In gas-engine blowers great fuel economies are made possible, as stated in the main paper.

DISCUSSION

E. T. CHILD (written). In Par. 7, Professor Trinks refers to the difficulty which exists with all poppet valves; namely, the fluttering of the valves and the impossibility of bringing them promptly and quietly against their seats at high speeds. The last sentence of this paragraph, which states that in ordinary American blowing-engine practice poppet valves close so near the dead center that for all practical purposes they may be considered as closing on time, without slipping back of air, is true only when poppet valve engines are run at very slow speeds.

In the many cases in which we have taken off heads equipped with poppet valves and replaced them with heads fitted with the Southwark sliding valves, the delivery of the engine has been increased from 50 to 100 per cent, through eliminating the loss due partly to

leakage and slip, and partly to the fact that the engines could run efficiently at much higher speeds than they had been used to running with the poppet valves.

In Par. 13 and in several subsequent places, Professor Trinks speaks of obtaining pressures in the air cylinder within ½ lb. of the atmosphere. With Southwark valves, we do much better than this. We assisted some years ago at a series of tests of blowing engines with Southwark valves at the Lackawanna Steel Company in Buffalo. The report on these tests was as follows: At mid-stroke the suction varies from 3 oz. per sq. in. at 45 r.p.m. to 5½ oz. per sq. in. at 80 r.p.m., but at the end of the admission stroke the blowing cylinder is filled with air which is always within 1½ oz. of atmospheric pressure.

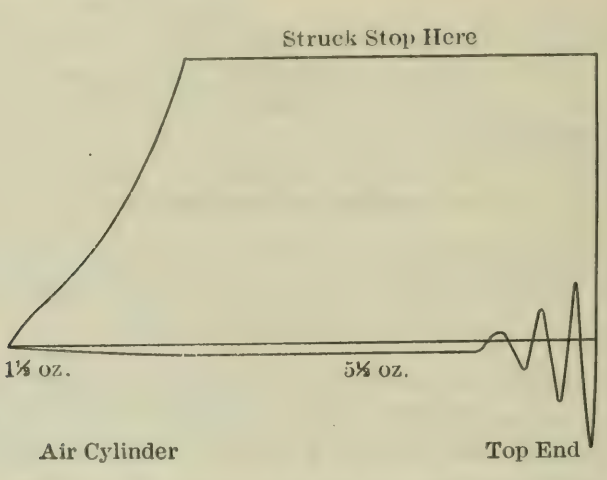


FIG. 34 TYPICAL INDICATOR CARD WITH SOUTHWARK VALVES

Fig. 34, shows a typical card taken during these tests. The amount given, 1½ oz., is looking at the matter from the worst point of view. It seems probable that the pressure in the cylinder at the end of the suction stroke is actually equal to atmospheric pressure, owing to the fact that the inertia of the column of air rushing through the inlet valve is enough to make up for the friction loss. This is the more likely from the fact that the Southwark inlet valve does not have harmonic motion, but remains wide open until almost the end of the stroke, when it is closed very rapidly by a cam. We are informed by Mr. Longacre, chief engineer of the Ingersoll-Rand Company, that he has frequently found that the inertia of the air entering through well designed inlet valves is more than sufficient to overcome the friction.

It will be noticed in Fig. 34 that a very light spring was used,

which does not show the top of the card at all, since the indicator struck the stop before reaching the pressure at which the blowing engine was delivering air. This light spring also accounts for the vibration at the beginning of the inlet stroke, which was doubtless due to the large inertia of the indicator parts as compared with the strength of the spring.

In connection with these tests at Buffalo, measurements were made by two methods to find out how much the air was superheated at the beginning of the compression stroke. This was found to be 4 deg. fahr. in the worst case and appeared to be due mainly to the contact of the air with the hot cylinder walls. The heating was a trifle less at high speeds than at low speeds, which shows that it was not due to friction through the inlet valve.

In Par. 20, the action of the Southwark valve gear is described and it is stated that the actuating cylinder is so proportioned that for a given speed of engine, a given blast of pressure and a given method of lubrication the valve opens at the correct instant. In modern Southwark gears, however, there is a check valve in the pipe leading air to the actuating cylinder, and this check valve is held shut by the pressure in the blast main. Air is therefore not admitted to the actuating cylinder until the pressure in the air cylinder becomes almost equal to that in the blast main. This arrangement overcomes the defect which Professor Trinks speaks of, of having the outlet valves open too soon. Also, our modern valves give considerably greater areas than those mentioned. They provide 20 per cent for the inlet and 15 per cent for the outlet whenever these large amounts appear to be desirable. If required, these amounts can be made 30 per cent and 20 per cent, respectively. We have not noted any falling off in the popularity of this valve gear. On the contrary, we have been busily engaged taking off heads of other design and substituting Southwark heads at many furnace plants in this and other countries.

The Southwark gear operates as satisfactorily at 600 ft. speed as at 300 ft. and avoids the losses due to tortuous passages mentioned by the author. Regarding the wear of the Southwark gear it will be noted that the valves do not move under pressure, but that at the moment when they are opened and shut, the pressure inside and outside the cylinder is balanced, causing the valves to float when moving. This doubtless accounts for the small amount of wear. It would not be possible to move the valves under these balanced conditions if they moved with harmonic motion, since they would then be moving practically all the time.

The performance of these valves is indicated by the record of 16 engines installed at the plant of the Lackawanna Steel Company, Buffalo, which have been running about eight years at 60 r.p.m. On horizontal gas engines in Europe, many engines run 80 revolutions continuously, night and day, and any speed less than 65 r.p.m. is abnormally slow. Of two engines 96 in. in diameter by 44 in. stroke, running 70 revolutions, one ran a total of 4475 hours out of a possible total of 4525; the other ran 4358 out of a possible total of 4452, after which one of the engines made a non-stop day and night run of over five months. Such results speak for themselves, and are not possible with engines having a large number of lifting valves, some of which are sure to need frequent attention.

We do not agree with Professor Trinks as to the quietness of the small poppet valves, since the impression we have received from all of the engines so equipped is that they were making a great deal of noise.

F. E. Cardullo thought that builders of blowing engines could follow the lead of the pumping engine manufacturers by using a large number of small valves of about 3 in. in diameter instead of large ones of 18 to 20 in. in diameter, which must be mechanically operated and are subject to unusual mechanical stresses and high temperatures. These large valves are more satisfactory with air than they would be with water, but objections are of the same character, and with high speeds are of the same validity as the objections to similar valves in water pumps. Suitable material should be used, perhaps steel plates, and with a very small rise almost no fluttering or vibration would be found. On account of the great simplicity of construction, such valves would give no trouble, and shutdowns from defective valves would be reduced to a minimum.

The Author. Fourteen years ago the writer saw a blowing engine with an almost countless number of small aluminum valves about $2\frac{1}{2}$ in. in diameter, all in strict compliance with the ideal of Professor Cardullo. Yet that type of engine did not survive, partly because the failure of even one valve of that size demands a shutdown for repair, just as it does with larger valves. The frequency of shutdowns is thereby increased and since it is impossible to tell definitely which valve has let go, hunting for the location of the broken valve is aggravating.

The use of a great number of small valves is inseparable from exces-

sive clearance space, a condition not harmful in water pumps, but impossible for blowing engines. Finally, the question of frictionless guiding is much more difficult for small valves than for large ones, because flexible members for small valves become too delicate to be practical.

The first paragraph of Mr. Child's discussion applies only to the high-lift, non-mechanical closed poppet valve, found in very few American blowing engines and not to the low-lift plate valves with frictionless guides.

It is true that poppet-valve cylinder heads have been replaced by Southwark heads and that great economies have been obtained by the change, but the poppet valves in these cylinders were of antiquated design. The writer does not know of a single instance of replacing modern low-lift multi-ported poppet valves by a Southwark gear. It should also be noted that in any case of replacement the old engine is usually worn out and in need of repair, whereas the new engine which takes its place is in first-class condition. As a proof of this statement the writer wishes to offer personal knowledge of two Southwark engines which, when new, furnished the wind for a furnace at 40 r.p.m. After a few years of operation these same engines had to run at 55 r.p.m. to furnish the same amount of wind.

While the writer values the receipt of the information that Southwark engineers have made an attempt to correct the harmful preopening of the discharge valve at slow speeds by the interposition of a check valve, he regrets that the information was not available at the time of preparing the paper.

Another point in Mr. Child's discussion needs attention. He states that Southwark valves do not move under pressure. If this were true they would have to move through the lap at an infinite velocity and no valve gear can be designed which is strong enough to accomplish this feat. Cam diagrams will show the untenability of his statement. It will be recognized that time for motion must be allowed and that a considerable amount of valve wear is unavoidable in order to save the operating cams from excessive wear or breaks.

No. 1322

POWER FORGING, WITH SPECIAL REFERENCE TO STEAM-HYDRAULIC FORGING PRESSES

By Barthold Gerdau, Düsseldorf, Germany
Non-Member

and

George Mesta, Pittsburgh, Pa.
Member of the Society

This paper deals principally with the most modern methods of producing forgings, and in order to appreciate these methods it will be necessary briefly to review what has been done.

2 All are acquainted with the fact that the power of the hand hammer is not derived from its weight alone, but from weight combined with velocity imparted by muscular power during a relatively long stroke. The same is true of the sledge and of all sorts of drop hammers, whether lifted by friction, water power or steam.

3 The power of drop hammers depends solely upon the mass of the hammer and its lift; whereas, in a certain kind of hammer operated by steam or compressed air additional power is obtained by increasing the acceleration beyond that caused by gravity alone.

4 While the steam hammer is older than the present generation, this can hardly be said of the modern type of forging press, illustrated diagrammatically in Fig. 1. A plunger *A* is forced by hydraulic pressure against the forging *B*, thus exerting a steady pressure instead of a blow. All forces are self-contained and no free forces are transmitted to the foundation.

5 In the course of events the forging press began to supplant large steam hammers everywhere, whereas the small hammer, operated by steam or compressed air, has held its own for small work. In the authors' opinion, the logical reason for this fact is the following: Small pieces cool quickly and, as a rule, require a finer surface finish than is demanded in large forgings. Since the hammer strikes

Presented at the Spring Meeting, Pittsburgh 1911, of The American Society of Mechanical Engineers.

a blow, the force of the impact increases as the forging cools, so that work is done on the surface of the forging, even when it is quite cool. The press, on the other hand, would be powerless to do any work on the cold piece, unless the anvil surface was reduced very much.

6 Thus the hammer will probably always be used for small work and for making tools, because tool steel must not be heated as much as machinery steel or wrought iron.

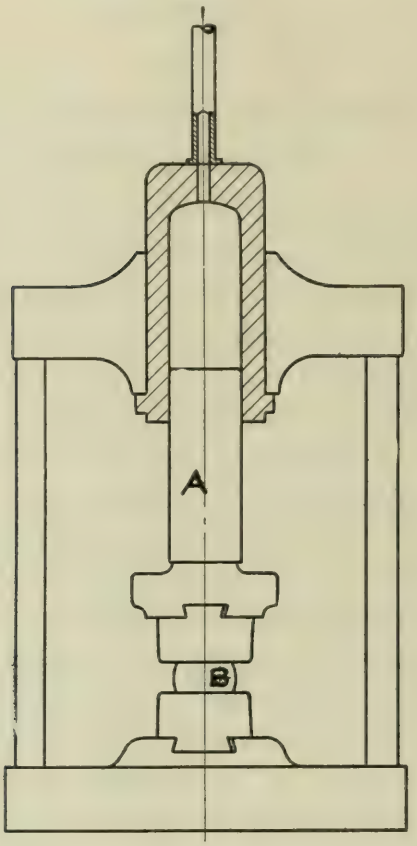


FIG. 1 SIMPLE TYPE OF FORGING PRESS

7 The very features which make the hammer desirable for small work make it undesirable for large work. The hammer acts upon the surface of the material, driving the surface over the core without compressing the latter, as in Fig. 2. The press, on the other hand, exerts a continuous pressure, which forces the semi-fluid material of the forging to flow under compression, as in Fig. 3, which process tends to increase the density of the material.

8 It is thus evident that from the standpoint of improving the quality of material of forgings, the press is superior to the hammer, and it would be considered even more useful if, under equal conditions, it used less steam than a hammer would consume.

9 To investigate the comparative steam consumption, a con-

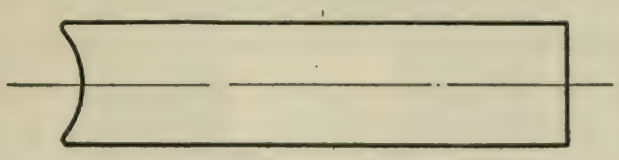


FIG. 2. SHOWING EFFECT OF HAMMERING

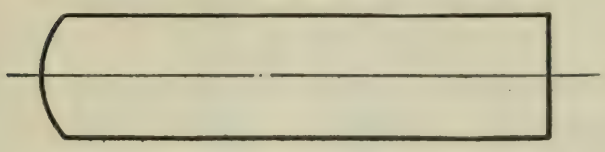


FIG. 3. SHOWING EFFECT OF FORGING BY PRESSURE

crete case will be assumed. Let the dimensions of a hammer with live steam above the piston be as follows:

Weights of falling parts $P = 5$ long tons
Diameter of steam cylinder $= 27\frac{1}{2}$ in.*
Stroke of piston $= 79$ in.
Initial steam pressure $= 107$ lb. per sq. in.
Mean effective pressure $= 71$ lb. per sq. in.
Face area of anvil $= 186$ sq. in.

Then we find :

a Mean effective accelerating force (P_2) due to steam pressure

$$= \frac{\pi}{4} \times 27.5^2 \times 71 = 42{,}000 \text{ lb.}$$

b Acceleration provided by P and P_2

$$= \frac{(42{,}000 + 11{,}200)\ 32.2}{11{,}000} = 155 \text{ ft. per sec.}$$

c Velocity (V) of moving parts at end of stroke

$$= \sqrt{2 \times \text{acceleration} \times \text{stroke}} = 45 \text{ ft. per sec.}$$

d Kinetic energy stored up in moving parts

$$= 53{,}200 \text{ lb.} \times \frac{79}{12} \text{ ft.} = 350{,}000 \text{ ft-lb.}$$

* These dimensions were converted from metric dimensions, which accounts for the odd sizes.

For hammers of this size and correct temperature of forging, it may be assumed that the moving masses are brought to rest within a space of $1\frac{5}{8}$ in.

e Basing the retardation of the moving parts upon this distance the average retarding force is found to be 2,580,000 lb., or approximately 1200 long tons. This figure may be termed the force of the blow and will be used below for comparison with a press. This force was based upon a retardation space of $1\frac{5}{8}$ in. It would be wrong to assume that all of this space is utilized for compressing the forging. The elasticity of the foundations absorbs a considerable amount of the work of the blow. A correct mathematical treatise of these losses meets with insurmountable difficulties. From the experience of the authors it appears that almost one-third of the work is lost in vibrations so that only about 1 in. of useful compression or deformation of the forging results.

f Since it is intended to investigate the relative economy of different methods of forging, the steam consumption per working stroke of the hammer in question will now be computed. Steam is needed for the lifting and for the down-stroke. The steam for the up-stroke may be figured approximately from the equation that pressure multiplied by change of volume equals work done. Thus cubic feet of steam required for up-stroke

$$= \frac{11{,}200 \times \frac{79}{12} \text{ ft-lb.}}{144 \times 107 \text{ lb. per sq. ft.}} = 4.8 \text{ cu. ft.} = 1.3 \text{ lb.}$$

10 For the down-stroke a cylinder full of steam at 71 lb. per sq. in. pressure may be taken, which will be sufficiently close for this purpose. Thus the volume of the steam per down-stroke

$$= \frac{\frac{\pi}{4} \times 27.5^2 \times 79}{144 \times 12} = 27 \text{ cu. ft.} = 5.4 \text{ lb.}$$

The steam consumption of one cycle then equals 6.7 lb.

11 This steam consumption is based upon correct operation of the hammer, which, however, is seldom realized if the operator neglects to close the throttle at the instant the hammer touches the forging, because in that case the pressure in the cylinder may rise to

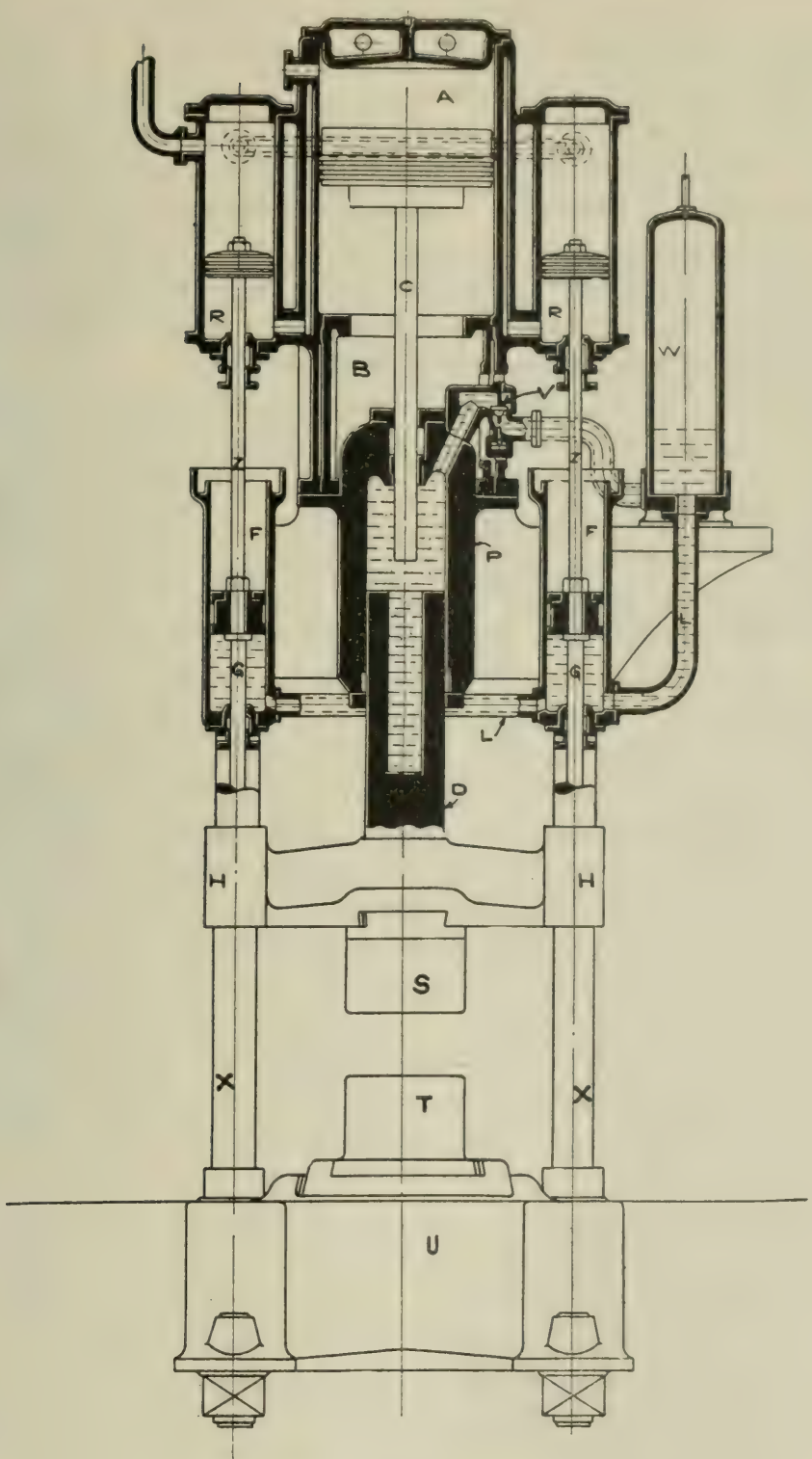


FIG. 4 STEAM-HYDRAULIC FORGING PRESS

boiler pressure. If the hammer is used to strike light blows, full strokes may be made, using the hammer principally as gravity drop, or short strokes may be made with live steam on top. In either case the steam consumption per unit of work done is enormously increased. The great number of combinations possible forbids giving figures.

12 Turning to the steam consumption of the press, we find that conditions are not so simple. It will be remembered from Fig. 1 that the press is operated by water under pressure. The steam consumption then depends first upon the method of producing the water pressure and second upon losses of water pressure between its production and its application to the press plunger.

13 Before the intended computation can be made, it is evidently necessary to review the various types of presses on the market. Two general types of presses exist: (*a*) The so-called purely hydraulic presses whose pressure-water is furnished by separate pumping engines; (*b*) the so-called steam-hydraulic presses in which water under pressure is generated in a steam intensifier.

14 The general type mentioned under (*a*) is best illustrated by Fig. 1; the pipe at the top of the cylinder is connected either directly or by means of a hydraulic accumulator to a high-pressure pump which may be either direct-acting or of the crank flywheel type.

15 The type mentioned under (*b*) is shown diagrammatically in Fig. 4. The press consists of a heavy base *U* and four uprights *XX*, supporting the hydraulic and steam cylinders above. *S* and *T* are the dies, the upper one being carried by the crosshead *H*. This crosshead is moved into position for forging by the use of the steam cylinders *RR*. Pressure for forging is obtained by admission of steam into the top of cylinder *A*, which moves the piston downward and forces rod *C* into the hydraulic cylinder *P*. This action again causes the plunger *D* to move downward, carrying with it the crosshead and the upper die. The effect is that of a steam-actuated intensifier.

16 Hydraulic cylinder *P*, balancing cylinders *FF* and about one-third of air chamber *W* are filled with water. Air pressure of about 100 lb. is pumped into air chamber *W*, which is sufficient to move piston rod *C* with its piston to the top of steam cylinder. The areas of plunger *D* and cylinders *FF* are equal, so that the downward force on plunger *D*, due to the air pressure on the water, is counterbalanced by cylinders *FF*. By opening check valve *V* in the connection between air chamber *W* and hydraulic cylinder *P*, thus allowing water to flow from the hydraulic cylinder to the air chamber,

and admitting steam to cylinders *RR*, the crosshead can be raised to any position and similarly it can be lowered.

17 Referring to type (*a*) it is found that the pumps commonly work against a pressure of 4300 lb. per sq. in. which is maintained by an accumulator. Unfortunately, the process of forging does not always require all of this pressure, but on the average only about 50 to 60 per cent of this amount, the remaining 40 to 50 per cent being lost in the controlling valve.

18 It might appear that it would be more economical to let the pump work directly against the press without an accumulator and even without intermediate controlling valves, but this scheme is too ideal to be practical. It has been found necessary to provide controlling valves, by-pass valves, throttling governors and other means for regulating the speed and power of the press.

19 Nevertheless, this type of press has one advantage, namely, that it can make a continuous full-pressure stroke of the maximum length for which the press was built. For this reason this type is most advantageous for drawing operations, for instance, the drawing of tubes, etc.

20 Taking up type (*b*) as illustrated by Fig. 4, it has already been mentioned that the pressure is generated by a direct-acting intensifier which is similar in its action to a direct-acting pump. While at first thought it might appear that the direct-acting intensifier, which necessarily has to maintain an almost constant steam pressure throughout the stroke, would be inferior to a high-class compound crank and flywheel pump, experience has taught that the opposite is the case, the reason being that the work is intermittent and requires a continually varying amount of power.

21 After these explanations we are ready to take up the computation of the steam consumption of a press for the same work which formed the basis for a similar calculation for the steam hammer. From what has been said it is evident that the type of pump, the conditions under which it works, the proper balancing of the accumulator pressure with regard to the work and other factors, affect the steam consumption. Therefore the following figures should be considered only as an attempt to solve the problem.

22 The effective work of the hammer was 240,000 ft-lb. per working stroke. Basing the press on the same work and taking the efficiency of the transmission pipe as 90 per cent, the pressure drop in pump ports as 10 per cent, and the efficiency of the pump as 75 per cent, the work actually to be done in the pump

cylinder is 395,000 ft-lb. Let the pump be driven by a steam engine whose steam consumption is 32 lb. per h.p-hr. on account of the intermittent working; then the steam required for the equivalent work is

$$\frac{395{,}000 \times 32}{33{,}000 \times 60} = 6.3 \text{ lb.}$$

This shows that the purely hydraulic press is slightly more economical than the steam hammer.

23 Matters are a great deal simpler for the direct-acting steam-hydraulic press. Basing the calculation upon the fact that pressure times change of volume equals work, and assuming a steam pressure of 150 lb. per sq. in. gage, we find the necessary volume of steam to be

$$\frac{240{,}000}{150 \times 144} = 11 \text{ cu. ft. or } 3.7 \text{ lb. of steam}$$

For the sake of simplicity friction of pistons has been neglected in all cases. This last result is reliable, whereas in the case of the purely hydraulic press and in the steam hammer, additional losses may occur which cannot very well be considered on account of their uncertainty. The excellent economy of the direct-acting steam hydraulic press presupposes that no dead volume of clearance space has to be filled before the motion of the steam piston against its full resistance begins.

24 The importance of this last sentence is so great that the authors take the liberty of repeating it in different words. The expression used actually includes three separate facts: (*a*) The steam piston of the intensifier must be in its starting position with practically no clearance; (*b*) the movable anvil must practically rest on the piece to be forged; (*c*) there must be no voids in the plunger-and-intensifier cylinder.

25 These three principles are embodied in the design shown in Fig. 4, which, in the authors' opinion, represents the most practical form which this type of press can assume. In this system the formation of voids in the water space is impossible, because the intensifier plunger floats upon the water at all times; in other words, the direction of motion of the intensifier piston and plunger is the same, which thus provides a continual force closure. If such force closure were absent, and in some types of steam hydraulic presses it actually is, there would be no guarantee of a filled water space and increase of steam consumption would be the inevitable result. Moreover, it will be seen from Fig. 4 that the pressure is generated directly

in the same cylinder in which it is used, thus avoiding the losses which must occur where the intensifier is separated from the press itself and the pressure water has to be forced through pipes and connections.

DISCUSSION

J. I. Rogers.[1] With regard to the assertion in Pars. 7 and 8, that hammer forgings have concave ends and that press forgings have convex ends, the shape of the ends really depends more on whether the apparatus used is heavy enough rather than on whether it is or is not a hammer. There are plenty of forgings made with heavy hammers that have convex ends. In the same way there are press forgings which are apt to have concave ones if the press is not powerful enough.

In Par. 18 direct-acting presses are referred to, which, acting directly from pumping engines using flywheels and expanding the steam, have been in very successful operation in this country in the making of guns and armor plate. They have given very satisfactory service, the only objection being the great cost of first installation.

In Par. 22, having figured that there is about 240,000 ft-lb. of work to be done on the forging by the hammer, Mr. Gerdau compares the press with the hammer, assuming that the work necessary to accomplish equal physical results on the forging is the same with both hammer and press. I believe from actual experience that this assumption is incorrect. The amount of work to be done on a forging depends upon the speed with which it is accomplished. That is, a hammer does not need to be quite so large as a press because it works at a different speed. On the same forging the actual force necessary to overcome the resistance of the metal will depend on the speed of the hammer or the press.

In some experiments on two 10,000-ton presses (two of the largest forging presses in the country), one of the direct-acting type and the other of the intensifier type, all conditions were made as nearly alike as possible in the different tests, the speed at which the press ran alone being changed. When the work was done in 50 seconds it required 10,000 tons; when in 32 seconds, only 6500 tons; and when in 17 seconds, only 4000 tons, this being the total pressure in each case. Therefore to do the same physical work on the hot forging, the force necessary depends upon the speed, that is, the speed with which the forging is compressed after the die first touches it.

[1] Consulting Engineer, 165 Broadway, New York.

Fig. 5 shows graphically the amount of the pressure or the force of the blow delivered by the pressure of the press or by the hammer. The ordinates represent the pressure of the press or the force of the blow of the hammer and the abscissae the distance the die moves in accomplishing the work. Taking this distance constant for means of comparison, the hammer will deliver the greatest blow at the beginning of the compression of the forging and have a curve like *H*. The press, on the other hand, will exert its greatest pressure at the end of the compression of the forging, starting with zero and gradually building it up as more is required, as in curve *A*. With a more slowly moving press it will run as curve *B*, and so on.

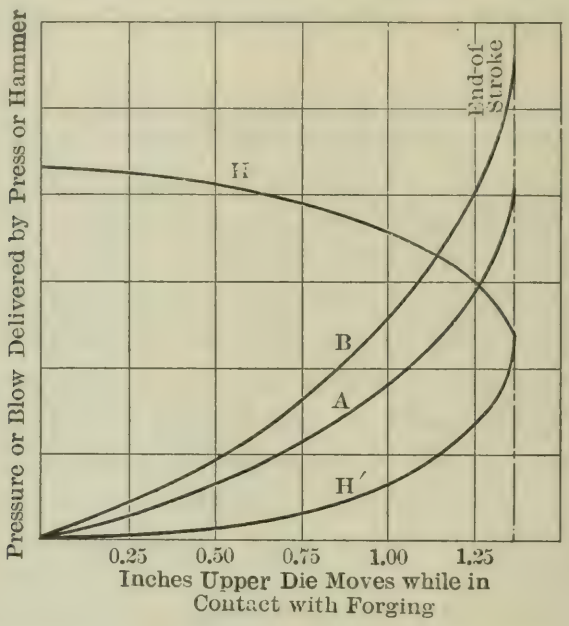


Fig. 5 Comparison of Blows Delivered by Press and by Hammer

In the case of the press, the resistance offered by the metal is at all times equal to the pressure exerted by the die; thus the areas under the curves *A* and *B* represent the work done in the two cases. In the case of the hammer the resistance offered by the metal is not equal to the force of the blow delivered until the end of the stroke. Thus the area under curve *H* does not represent the work done by the hammer, this being represented by the area under a curve *H′* which represents the resistance offered by the metal to the blow of the hammer. Accordingly, we cannot say that the amount of work to accomplish the same result on a forging is the same in all cases. In addition to the speed of forging, the force necessary to overcome the resist-

ance of the metal depends upon so many things, such as the temperature of the forging, the composition of the metal, the shape of the dies and the condition of their faces, that it is a very difficult quantity to figure accurately.

I believe that the press and the hammer have each their own fields. In the selection of either the question of steam consumption should be one of the minor factors, the most important requisite being to select the tool that will make the best forging in the most efficient way, taking into consideration the speed of production, the facilities for changing dies, etc., and that will not be out of service under repairs. Each case must stand upon its own requirements for the particular work in hand.

W. E. Hall. Referring to the computation of steam consumption of steam hammers, this type of hammer has the reputation of being a steam eater, but the method used by the authors gives results which seem excessive. Efforts have been made for several years, but without success, to have placed before the Society some comprehensive information relative to their steam consumption, etc. The only records available, so far as we know, are some indicator cards (Figs. 6 and 7) taken by the writer some 21 years ago, which are quite incomplete, but do throw some light upon the steam consumption.

The hammer was built by William Sellers & Company, Inc., and the nominal rating was 2 tons, the stroke $35\frac{3}{8}$ in. and the indicator spring 40 lb. The hammer was not available for measuring the cylinder diameter until some weeks after the cards were taken. This dimension (the cylinder having been re-bored one or more times after installation) was subsequently accidently lost. This and other matters prevented a completion of the investigation. The steam for the hammers was obtained from boilers located over the smith shop heating furnaces and the variation of the work in the furnaces produced rather abnormal differences in the boiler pressures for the different cards. The hammer was located about 50 ft. from the boilers and was operated in the usual way by a hammer man under the guidance of the foreman.

The cards are presented because they show from the cut-off, etc., that the basis used by the author for computing the steam consumption would seem to give an abnormal result. They contain some rather interesting features such as the effect of cushioning, etc., and it is rather unfortunate that there seems to be no record showing the most economical method of distribution and the most efficient way of operating such a widely used shop appliance.

THE AUTHORS. Mr. Rogers' contention is correct, that hammers which are very heavy in proportion to the size of the forging will produce convex ends. The size of hammers, however, has not kept pace with that of forgings, which fact accounts for the concave ends on large forgings made by hammers. The vibrations of the ground from hammers large enough for heavy forgings are so serious that the use of a press is imperative. A heavy hammer is better than a light one because the former owes its power more to its mass and less to its velocity. The action during contact is slower than that of the light hammer and therefore approaches the action of the press.

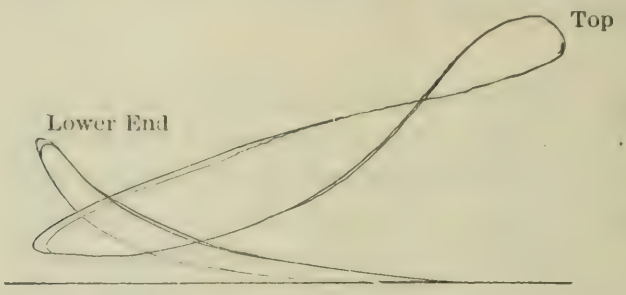


FIG. 6 EFFECT OF SHORT STROKE NOT STRIKING

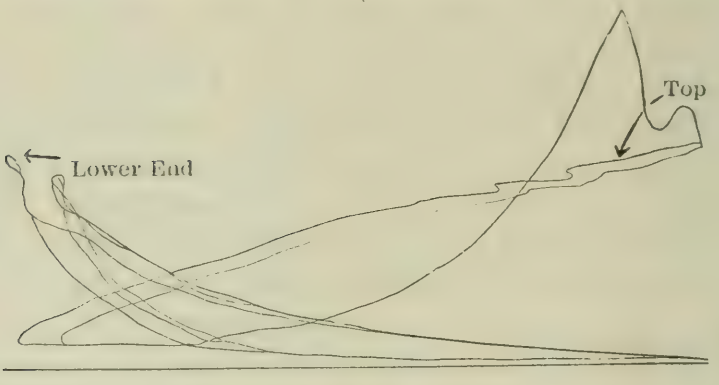


FIG. 7 EFFECT OF LONG STROKE STRIKING HARD

We do not understand the inference made by Mr. Rogers in the second paragraph of his discussion. We have never denied that presses operated by crank and flywheel pumps are in successful operation, but have stated that the direct-steam hydraulic press is for certain processes not only cheaper, but also more economical.

We agree with Mr. Rogers that the work required to shape a forging depends upon the time consumed, but cannot assent to the statement that a hammer works faster than a press. To the casual observer it might appear that the hammer moves faster, but while it does

so, most of its fast motion is used for getting ready to do work. The press consumes practically no time in getting ready, and works all the time. It may be interesting to note that the forge shop of the Mesta Machine Company has been able to cut the time on forgings considerably since the hammer was replaced by the press.

We do not wish to discuss the correctness or incorrectness of Fig. 5, because that would involve a theory of plasticity, coupled with dynamics of the anvil and ground or soil. While our experience leads us to believe that these curves are not right, we cannot disprove them for want of tangible data.

Referring to Mr. Hall's discussion, it appears to us that the cut-off like shape of the indicator cards misled him. Both cards show a great deal of negative work, and when this is substracted from the positive work the high steam consumption per unit of work becomes apparent.

No. 1323

MEETINGS SEPTEMBER-DECEMBER

MEETINGS PREVIOUS TO THE ANNUAL MEETING

ST. LOUIS, SEPTEMBER 20

Meeting with the Engineers Club of St. Louis. The Colorado Spring Water Works: Paper by Hiram Phillips, member Engineers Club and the American Society of Civil Engineers.

NEW YORK, OCTOBER 9

Factory Construction and Arrangement: Paper by L. P. Alford and H. C. Farrell. The paper describes the arrangement and construction of the reinforced-concrete factory buildings of the United Shoe Machinery Company, at Beverly, Mass., with reference to their adaptability to the manufacture of light machinery, and discusses the advantages and disadvantages of concrete floors, the one-shop plan, the artificial lighting installation, and various modifications adopted by the company from their experience. Published in The Journal, October 1911. For the discussion see The Journal, March 1912.

SAN FRANCISCO, OCTOBER 17

Dinner in honor of Col. E. D. Meier, President of the Society. Discussion of Society matters and appointment of committee to confer with representatives of other engineering organizations with regard to engineering congress proposed for 1915.

BOSTON, OCTOBER 18

Power System of the Pacific Mills: Paper by F. A. Wallace. The paper gives a description of the plant, transmission system and motor drives, methods and organization for operation and inspection, and considerable information as to costs. Published in abstract in The Journal, March 1912.

PHILADELPHIA, OCTOBER 18

Meeting with the Franklin Institute. The Practical Application of Scientific Management to Railway Operation: Paper by Wilson E. Symons, consulting engineer, Chicago, Ill. Many statistics, covering employees' compensation, operating revenues and expenditures, division of expenditures and balance sheet, and efficiency, were given to show that the economic operation of railways had already been reduced to a minimum and that scientific management could not produce further saving.

SAN FRANCISCO, NOVEMBER 2

Meeting to consider the engineering congress to be held in 1915.

ST. LOUIS, NOVEMBER 11

Business meeting to consider prospective applicants and other routine matters.

NEW YORK, NOVEMBER 11

Meeting on Welding : Papers,Modern Welding Processes, H. R. Cobleigh: Thermit Welding, G. E. Pellissier; Arc Welding, C. B. Auel, assistant manager of works, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa. The first of these papers gave a general account of the processes employed in welding and the progress made by them, laying considerable stress on the flame process; while the others dealt with the thermit and electric processes respectively. Papers and discussion published in The Journal, January 1912.

NEW HAVEN, NOVEMBER 15

Meeting on Cost of Power with Small Units of Various Types: Papers, A Small Producer Gas Power Plant in a Wood-Working Shop, Albert W. Honywill, Jr., giving the cost of operation and working conditions in a producer plant installed at the Lampson Lumber Company, New Haven; Comparative Cost of Power by an Oil Engine and a Steam Engine, E. H. Lockwood and F. P. Pfleghar, president Pfleghar Hardware Specialty Company, New Haven, giving data on the subject; The Status of Small Steam Turbines, W. J. A. London, giving interesting comparisons of turbine-driven and engine-driven pumps for boiler-feed purposes, obtained from tests on scout cruisers, Birmingham, Chester and Salem. At the evening session, an address on the work of engineering societies

was made by Col. E. D. Meier, and Prof. C. F. Scott presented a paper on The History and Development of the Hartford Electric Light Company, describing its central station service. An account of the meeting was published in The Journal, March 1912.

BOSTON, NOVEMBER 16

Some Refractory Substitutes for Wood: Paper by Charles L. Norton, outlining the various attempts made by different scientists to produce some satisfactory substitute and describing an invention of his own. An account of this meeting appears in The Journal, March 1912.

THE ANNUAL MEETING

The thirty-second Annual Meeting of the Society, one of the largest and most enthusiastic of its history, was held at the Society Headquarters in New York, December 5–8, 1911. The feature of this year's meeting was the large average attendance, the total registration exceeding 1200, with 687 members and 545 guests in attendance at its various gatherings. The Committee on Meetings had prepared a program of unusual interest, comprising contributions from three of the sub-committees recently appointed to investigate special subjects, and all the sessions were well attended and brought out discussion of value. On Wednesday evening the address by Dr. Robert S. Woodward, of the Carnegie Institution of Washington, was much enjoyed by all in attendance, and appreciation was expressed of Dr. Woodward's courtesy in consenting to address the membership. Interesting and varied technical excursions and entertainment features arranged by the Committee on Meetings in New York, assisted by the Ladies' Reception Committee, formed a social background to the meeting and contributed to the spirit of fellowship which prevailed.

PROGRAM

Tuesday Evening, December 5

President's Address: The Engineer and the Future, E. D. Meier.

Report of Tellers of Election of Officers, and introduction of the President-Elect.

President's Reception in the rooms of the Society.

Wednesday Morning, December 6

Business meeting. Reports of the Council, tellers of election of membership, standing and special committees. Amendment to C-57, relating to the financing of geographical and professional sessions. Announcement of sub-committees of the Committee on Meetings. New business.

PROFESSIONAL SESSION

THE TURRET EQUATORIAL TELESCOPE, James Hartness.

Discussed by Ambrose Swasey, Wm. Kent, F. R. Hutton, Worcester R. Warner, Henry Hess.

EXPENSE BURDEN: ITS INCIDENCE AND DISTRIBUTION, Sterling H. Bunnell.

Discussed by H. F. Stimpson, Benj. A. Franklin, H. K. Hathaway, Wm. Kent, Harrington Emerson.

STANDARD CROSS-SECTIONS, H. deB. Parsons.

Discussed by F. De.R Furman.

Wednesday Afternoon

PROFESSIONAL SESSION

TESTS OF LARGE BOILERS AT THE DETROIT EDISON COMPANY, D. S. Jacobus.

Discussed by R. H. Rice, R. D. DeWolf, H. O. Pond, Wm. D. Ennis, H. deB. Parsons, H. H. Esselstyn, E. G. Bailey, Wm. Kent, R. C. Carpenter, E. J. Billings, J. W. Thomas, A. Bement, W. F. M. Goss.

STRAIN MEASUREMENTS OF SOME STEAM BOILERS UNDER HYDROSTATIC PRESSURES, James E. Howard.

Discussed by Francis B. Allen.

HERRINGBONE GEARS, Percy C. Day.

Discussed by F. E. Rogers, Wm. B. Burlingame, F. De R. Furman, W. C. Brown.

SIMULTANEOUS SESSION ON CEMENT MANUFACTURE

TOPICAL DISCUSSION ON CEMENT MANUFACTURE: The Dust Problem in Portland Cement Plants and its Solution, Otto Schott; Confirmation of the Advantages of Electricity to the Cement Manu-

facturers, J. B. Porter; Electrical Power in Cement Plants, F. H. Lewis; Protection, of Laborers from Accidents and Injury to Health in Cement Plants, Otto Schott; Methods and Appliances for Prevention of Accidents in Cement Plants, J. G. Bergquist; Depreciation and Obsolescence in Portland Cement Plants, Holger Struckmann.

Reception

Reception to members and guests of the Society, given by the Ladies' Committee in the rooms of the Society. Music and refreshments.

Wednesday Evening

Geo-Dynamics, or the Mechanics of the Formation of Worlds: Address by Dr. Robert Simpson Woodward, President of the Carnegie Institution of Washington, D. C.

Thursday Morning, December 7

Professional Session

The Core Room: its Equipment and Management, Henry M. Lane.

Discussed by B. D. Fuller, A. N. Kelley, A. E. Outerbridge, H. A. Becker, Thomas D. West, Richard Moldenke.

Tests of a Sand-Blasting Machine, Wm. T. Magruder.

Discussed by F. C. Brooksbank, W. S. Giele, A. G. Warren, J. M. Betton, S. C. Smith.

Die Castings, Amasa Trowbridge.

No discussion.

Variable-Speed Power Transmission, G. H. Barrus and C. M. Manly.

No discussion.

Simultaneous Session, Gas Power Section

Oil Engines, H. R. Setz.

Test of an 85-h. p. Oil Engine, Forrest M. Towl.

Discussed by C. E. Sargent, A. J. Frith, Wm. Sangster, Louis K. Doelling, C. W. Baker, L. B. Lent, H. J. Freyn, E. D. Dreyfus.

Design Constants for Small Gasolene Engines, Wm. D. Ennis.

No discussion.

Economy of 1000-Kw. Natural Gas Engine, E. D. Dreyfus and V. J. Hultquist.

Discussed by W. D. Ennis, W. S. McClelland, H. J. Freyn, John Willis, Howell C. Cooper.

Thursday Afternoon

Inspection of the White Star S. S. Olympic.

Thursday Evening

Reunion of the membership of New York, in honor of the newly-elected officers and visiting members, their ladies and guests, at the Hotel Astor. Dancing and refreshments.

Friday Morning, December 8

Professional Session

The Development of the Textile Industries of the United States, Frank W. Reynolds.

Discussed by Chas. T. Main, Geo. R. Stetson, John Eccles, G. H. Perkins, Chas. F. Scott, A. J. Herschman, Jas. A. Pratt, J. I. Lyle, C. W. Rice, Geo. M. Brill, Dwight Seabury, Mr. Williston, E. D. Dreyfus, Albert L. Pearson, Chas. T. Plunkett.

Rational Psychrometric Formulae: their Relation to the Problems of Meteorology and of Air Conditioning, Willis H. Carrier.

Discussed by R. C. Carpenter, L. S. Marks, O. P. Hood, G. A. Goodenough, R. C. H. Heck, Thos. M. Gunn.

Air-Conditioning Apparatus, Willis H. Carrier and Frank L. Busey.

Discussed by R. C. Carpenter, F. R. Still, J. I. Lyle.

Some Experiences with the Pitot Tube on High and Low Air Velocities, Frank H. Kneeland.

Discussed by G. F. Gebhardt, R. C. Carpenter, Willis H. Carrier, D. W. Taylor.

Friday Afternoon

Excursions to Thomas A. Edison Laboratory, Orange, N. J., Brooklyn Navy Yard, Brooklyn, Bush Terminal Company, Brooklyn, E. W. Bliss Company, Brooklyn, Ward Bread Company, New York City.

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C. E. LUCKE
H. DEB. PARSONS
W. E. HALL
C. J. H. WOODBURY

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F. A. WALDRON, *Secy. and Treas.*

F. H. COLVIN
EDW. VAN WINKLE
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ACQUAINTANCESHIP COMMITTEE

ROY V. WRIGHT, *Chairman*

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Afternoon

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LAWRENCE ADDICKS
F. E. EBERHARDT
R. E. FOX, JR.
H. L. GANTT
J. D. MAGUIRE
E. J. PRINDLE
F. E. ROGERS

Evening

L. A. SHEPARD, *Sub-Chairman*
H. R. COBLEIGH
W. C. DOUGLASS
C. I. EARLL
H. S. HAYWARD, JR.
F. R. HUTTON
M. C. MAXWELL
C. W. OBERT
C. F. SCOTT

Wednesday

Morning

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L. D. BURLINGAME
W. R. DUNN
M. P. FILLINGHAM
T. M. KEITH
H. P. MERRIAM
F. J. MILLER
FITZ-WILLIAM SARGENT

Afternoon

F. H. STILLMAN, *Sub-Chairman*
H. H. BARNES, JR.
H. D. GORDON
P. C. IDELL
J. P. ILSLEY
A. C. JACKSON
H. F. J. PORTER
W. S. TIMMIS

Thursday Morning

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A. FALKENAU
J. W. LIEB, JR.
W. M. MCFARLAND
C. H. PARSON
H. B. PROUT
E. A. STILLMAN
L. A. WHITCOMB

Friday Morning

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F. A. HAUGHTON
J. W. NELSON
N. B. PAYNE
W. W. RICKER
J. M. B. SCHEELE
M. M. UPSON
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No. 1324

THE ENGINEER AND THE FUTURE

PRESIDENTIAL ADDRESS 1911

By E. D. Meier, New York

President of the Society

If we could plot the progress of engineering in the last century in a plane curve culminating in the present, at the intersection of our axes the future would probably trace a line ascending in a great parabola.

But as we look back we can conceive of no equation which could express the achievements of the past in a single line, no matter how grand its sweep. The three dimensions which limit our knowledge of space are requisite to compass the varied activities of the engineer.

A century ago the distinction between civil and military engineer sufficed, but a few decades ago it became necessary to differentiate in turn the mechanical and the electrical engineer, while quite recently upwards of a hundred specialties were enumerated in the attempt to define the activities of the profession, each of which has its recognized experts. These are developed to fulfil the imperative demands in every art and industry for a greater refinement, precision and certainty as to the quantity and quality of the product.

Slowly but surely the superstitions and traditions which have so long encumbered our social life and hampered our free development, are exposed and annihilated by the altruistic labors of men who give their life to science. These are the high priests of the new dispensation. It is the duty, the glorious privilege of the engineer to receive their discoveries with reverent hands, and apply them to the solution of the practical problems of life.

Both these types of men are essentially modern products of an evolution which counts not by centuries, but by ages. Before the

An address delivered at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

first Cain of the stone age could appropriate the fruits of his brother Abel's labor there was a mechanic who fashioned his stone axe as patiently as he had lashed the pole to the curved branch with which Abel ploughed. The lame blacksmith who hammered out the greaves of Achilles and chiseled a whole panorama of barbarous deeds on his bossy shield, stood in such high repute among the warlike Greeks that they voted him a place among that rather disreputable coterie which ruled the small world of the day from Mount Olympus. Some centuries later there were a Democritus, a Bion and a Euclid, who developed geometry. But the clumsy method of numerical notation and the absence of algebra made the application to practical problems almost impossible. And the unfortunate habit of the noblest minds among the ancients to lay more weight on methods of reasoning and theoretical speculations than on the facts on which these should be based, retarded the union of science and practice. Even the inventive genius of Archimedes was hampered by these unfavorable conditions.

Those great road and bridge builders, the Romans, produced military engineers, but theirs were mainly static problems; and even their much vaunted aqueducts show lack of coöperation between science and practice. They were carried over valleys on costly structures inviting diversion or destruction at the hands of the enemy. With their excellent cement and their knowledge that water always seeks its level, their engineers might have built subterranean conduits.

The great engineer of the Renaissance, Leonardo da Vinci, with prophetic prescience conceived in the sixteenth century projects which the nineteenth and twentieth were to carry out in the light of scientific facts entirely unsuspected in his day.

The seventeenth and eighteenth centuries were engulfed by wars predatory and dynastic, and even the fierce upheaval of the French Revolution and the drastic destruction of feudalism by Napoleon left alive modes of thought based on an exaggerated reverence for the philosophy of Greece and Rome.

Early in the nineteenth century the scientific method came into vogue, and henceforth problems were studied and defined before their solution was attempted, and more intellectual labor expended in ascertaining facts than on reasoning about them. Thus the union between the mechanic and artificer and the student of nature's eternal laws became possible and permanent, and engineering developed from an art into a profession.

The men who spanned the Hellespont for the hosts of Xerxes, those who dug the irrigation canals which made Mesopotamia the granary of the ancient world, those who designed and built the engines with which Alexander battered the walls of Persian strongholds, those who cumulated the puny muscular force of thousands of Egyptian slaves for the herculean task of raising the pyramids, all these men were giants in their day; but now their very names are forgotten. The literature of that day and for long afterward was concerned mainly with kings and conquerors. Jurisprudence and medicine shared in some slight degree the attention and prestige which were almost wholly absorbed by war, and can hence trace their history back to the ancient world. Democritus was one of the few ancients with scientific bias; knowing his surroundings we can understand why he became the "laughing philosopher."

Engineering is the profession of the present, and will dominate the future.

Laws have been made by men ever since the family expanded into the clan or tribe. They naturally reflect the ethical standard of the average mind of the period. Far in advance of them are the precepts laid down by those who founded the great religions of the world And as we reverently discover and apply natural laws, we find new reasons and supports for these fundamental ethical conceptions.

The engineer then is a devout believer in natural laws. He knows that they are immutable and permit no exceptions. He needs no Supreme Court to define them as reasonable. They are the very foundation of the universe, and reason itself owes its existence to them. Every infraction of them brings its own punishment. The knowledge that every mistake or neglect inevitably results in failure is ingrained in the very fiber of his being. The vile doctrine evolved in the dark ages, "the king can do no wrong," which still causes occasional lapses of justice, has no meaning for the engineer. To tell the truth is not merely laudable and salutary, it is essential and necessary. To lie is not only wicked, it is ineffectual, absurd and ridiculous.

To men thus trained, the future of the race is to be confided. They are not to be merely learned men; they must possess knowledge. Those fundamental sciences which observe and explain the interrelation of matter and force, which weigh the distant planets and measure the wave lengths of sound and of light, must be the absorbing objects of their nightly vigils and their all-satisfying reward.

The savage hated work, and even in those golden periods, praised by romancers masquerading as historians, labor was despised as the doom of the slave. As progress demanded more and higher types of labor, various devices were invented to secure it. From the glass beads and the brass bangles of the Hottentot to their counterparts in polished stone and burnished gold in civilized communities, these devices were effectual; but the highest type of labor has never been purchased by such crude bribes. The enlightened man loves his work and finds in it his supreme incentive.

To a Copernicus or a Newton, a Watt or a Corliss, an Ericsson or a Fritz, an Edison or a Steinmetz, the ransom of a king would seem trivial compared with the satisfaction of knowing that he has given to his fellow men an achievement which marks a forward step in the evolution which will finally make us a race of rational beings.

When the "missing link" stood erect, walked and essayed articulate speech, this evolution began. The first man had crude but strong desires, and was a strenuous individualist. The predatory instinct was predominant, and the success of the family, the clan, the tribe, and finally the nation, depended on the potent warrior at its head. Laws and customs modified this predatory individualism in each community, but between neighboring communities there still held the rule, that

> "He shall take who has the power,
> And he shall hold who can."

The right of the mailed fist is still occasionally invoked by great nations, even though the individual citizens have been tamed and domesticated.

Commerce, which has in large measure wrested the control of the world from the war lords, has always had as its basic principle the rule, "Buy as cheap as you can, sell as dear as you can."

This has worked fairly well when cargoes of grain, wool, cotton, hemp, etc., were concerned. But where this principle was ruthlessly applied to the great producing industries of the world, where the comfort of the worker, the maintenance of his family, the very existence of his helpless children were in jeopardy, it has caused havoc, bred discontent and fomented revolution. No one who has worked among the contented, intelligent mechanics of a half century ago can view without distress and indignation conditions as they exist today. Labor unionism is a protest, dangerously near a rebellion, but not a cure.

When the commanders of the industrial soldiery wrought in their midst, understood their problems, solved their perplexities, aroused and shared their enthusiasm for the quality of the product there grew up an esprit de corps which is now sadly missing.

Without faith in the excellence of the goods produced, without enthusiasm in the work and in the leaders, productive labor becomes mere drudgery. The remedy lies in placing engineers in all the responsible positions in these great industries. Their special training fits them for leadership in this host; and leadership by him who knows, and who sinks himself in his work, always has and always will command that joyous and fervent support from his followers which money cannot buy.

Where nepotism is banished, and ability and perseverance are recognized, there great success is attained. Napoleon's army, the Carnegie Steel Works, the Pennsylvania Railroad, exemplify this. The increasing multitudes of special industries are literally hungering for engineers equipped with character, knowledge and devotion, to become the expert leaders.

Labor wars will cease when such men are given the power to provide that the share in the rewards of industry shall bear just ratio to the service rendered the community. There is a sane middle ground between grasping individualism and utopian socialism.

The forces and materials we employ in our manufactures have long been and still are the subjects of most careful analysis and experimentation, to find their precise load capacity and endurance. The study of the living force and of that highest and most costly material inherent in the workman has but recently begun. But already several promising theories are undergoing exhaustive tests on a large scale. Psychological study prescribes a humane basis for them all as the condition of success.

The unrest in the modern world has its basis in an underlying sense of injustice. The growing sense of community of interest, the knowledge of our dependence on each other, the ever-expanding humanitarianism, are all founded on scientific facts, and are becoming world movements. They fervently and emphatically answer Cain's question: "Thou art thy brother's keeper."

The engineer is responsible for the vast increase in appliances to meet every demand of that most voracious of living beings, man. The mass of mankind needs to be educated to understand and use them properly. He is in honor bound to supply this education; and as the crude dangers and fears of the earlier centuries vanished, so

the prejudices and superstitions of the Dark Ages must be swept away.

If our future professional brethren do their duty, and we know they will, the golden rule will be put in practice through the slide rule of the engineer.

No. 1325

THE TURRET EQUATORIAL TELESCOPE

A NEW ASTRONOMICAL OBSERVATORY

By James Hartness, Springfield, Vt.

Member of the Society

The subject of this paper cannot be properly considered without acknowledgment of indebtedness to at least three of our members, each one preëminent in one or more branches of astronomical science. Notwithstanding their great variety of interests, these men have found time to write long letters of advice and criticism in response to questions or on submission of drawings suggesting the various steps in the evolution of the present scheme. Years ago the writer thought he was especially favored in this respect, but later observation has proved that the same earnest, personal consideration is given to any one. It was from these men that the microbe of work in the astronomical field was taken, and it is a pleasure to acknowledge indebtedness to our Honorary Member John A. Brashear, and to Past-Presidents Worcester R. Warner and Ambrose Swasey, for advice and criticism of the optical and general features of the various schemes out of which this new observatory has evolved. Indebtedness is also acknowledged to James B. McDowell and other members of the staff of The John A. Brashear Co., Ltd., and to Alvan Clarke Son Corporation for painstaking care in answering questions and giving advice regarding the optical parts.

The new observatory grew out of an attempt to make an observatory in which the observer could work in comfort, independent of the outside temperature. To accomplish this end without serious handicap to good vision and instrumental precision involved a departure from previous forms.

2 For the purpose of defining the relation of the new observatory to preceding types it is necessary to refer to diagrammatic illustrations, Figs. 1A, 2A and 3A. These sketches are purely schematic and are not true to important details.

3 Fig. 2A for the purpose of this paper will be referred to as the Standard Observatory, since it represents a type most highly developed by reason of the greater number of instruments of this kind

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

that have been made. This instrument is not only the most efficient optically, but is probably the most reliable in point of mechanical precision when compared with any other instruments designed for covering the whole heavens and for general purposes. It has, however, the one serious handicap of requiring within the building a temperature equal to that of the outside air. It is for the purpose of overcoming this handicap that the instrument to be described was designed.

4 Fig. 1A is known as the Loewy Equatorial Coudé. This instrument was designed to shelter the observer in comfortable quarters.

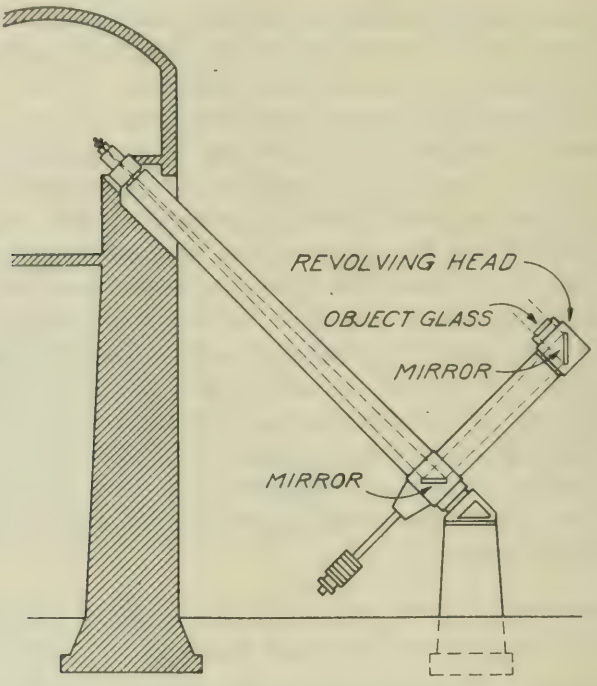


FIG. 1A DIAGRAMMATIC SKETCH OF THE LOEWY EQUATORIAL COUDÉ
Object glass, 23 in. in diameter with 25:1 focal length; largest diameter of larger mirror, 32 in. largest diameter of mirror at elbow, 23 in.

It is of French origin and a number of these are in use in Paris and elsewhere.

5 Fig. 3A is a schematic sketch of the new instrument which we propose to designate as the Turret Equatorial.

6 All of these instruments are refractors, using an object glass for collecting the rays of light and delivering them to a focal point within reach of the eyepiece.

7 In the standard equatorial the optical parts consist merely of the object glass and the ocular; for this reason it has the highest efficiency of any instrument for this purpose.

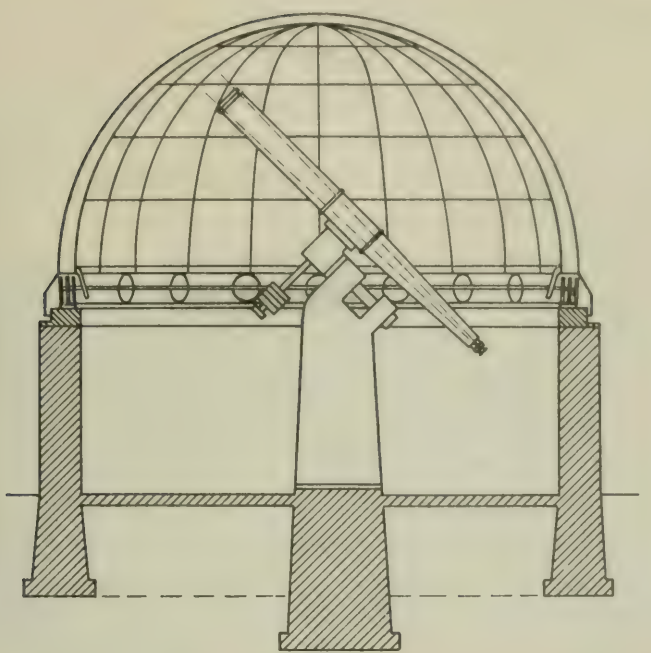


FIG. 2A STANDARD EQUATORIAL
Object glass, 10 in. in diameter with 15:1 focal length.

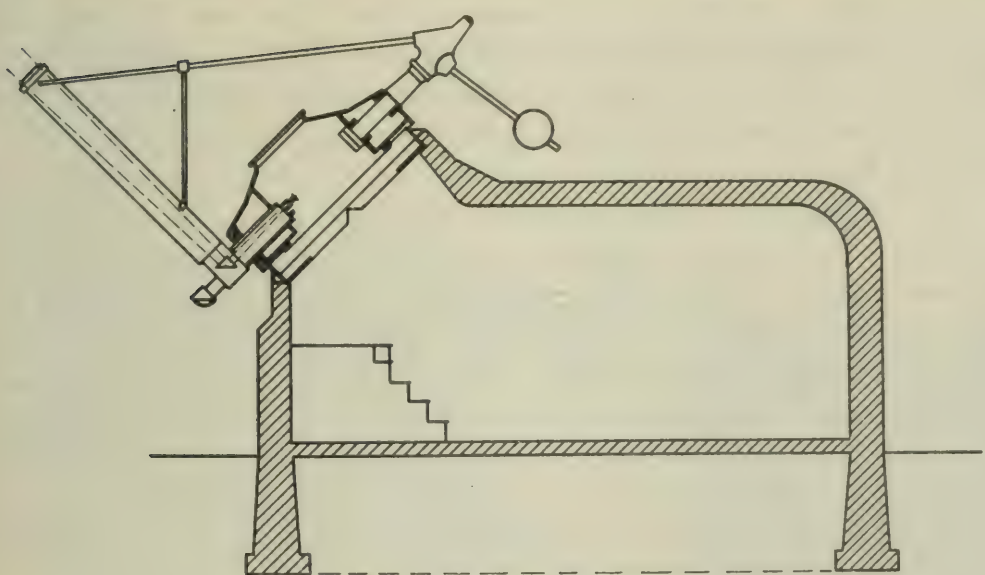


FIG. 3A TURRET EQUATORIAL
Object glass, 10 in. in diameter with 15:1 focal length, 2¾-in. prism at bend, which is approximately ⅓ the distance from eyepiece to objective. Actual size of beam of light is 2¼ in. at this point, or less than $\frac{1}{16}$ the area of objective.

8 The equatorial coudé offers comfort to the observer at the expense of a more or less serious optical loss, for between the object glass and the eyepiece two diagonal plane mirrors are interposed. In this instrument the beam of light comes to the eye at about the same angle as from a microscope on a table. The mirror at the object glass, of course, must be large enough to deliver the full bundle of rays, and since it stands at an angle of 45 deg., its major diameter must exceed the diameter of the object glass by about 40 per cent. This is also true of the diagonal plane at the elbow, although the diameter of the cone of rays is about 0.6 the diameter of the object glass. Notwithstanding this handicap, the equatorial coudé has doubtless been considered the best instrument which could be used by an observer comfortably housed.

9 There is another instrument, not shown in the sketches, somewhat similar to the equatorial coudé. It is jointed at the bend in the elbow, and avoids the necessity of the large mirror at the objective. It has not been included in this description because it does not cover the whole heavens. Its building obscures the circumpolar stars and that part of the heavens north of the building.

10 Mention should also be made of the Tower telescope at Mount Wilson, Cal., designed for solar work. In this the beam of light is delivered downward by means of two reflectors through the object glass also located at the top of the tower. Reference might also be made to the various horizontal telescopes that have been used. But these instruments have thus far been used for only a limited part of the range covered by the standard equatorial.

11 In addition to the standard equatorial and equatorial coudé, both of which should be classed as refractors, special mention should be made of the Commons reflector as it is now mounted at Harvard University. This instrument delivers the beam of light to the observer, who may be seated at a desk, just as in the equatorial coudé observatory. It is, however, a reflector, and should perhaps be kept in a class by itself. It also uses two auxiliary mirrors to deliver into the eyepiece, and furthermore, like the jointed equatorial coudé, its working range is limited by the obscuring of part of the heavens by its own structure.

12 In all of the instruments designed for the comfort of the observer it has been necessary to introduce one or more reflectors, and barring the Commons reflector at Harvard and the jointed equatorial coudé, these auxiliary reflectors have been located near the objective.

13 In the present instrument the direction of the beam of light

FIG. A VIEW FROM A POINT SOUTHEAST OF THE OBSERVATORY
Note the various positions of the turret on its axis and the positions of the tube in Figs. A–G

FIG. B VIEW FROM A POINT ABOUT NORTHEAST OF THE OBSERVATORY

FIG. C VIEW SHOWING WEST SIDE AND SOUTH END

FIG. D WINTER VIEW TAKEN JUST AFTER COMPLETION OF GRADING, SHOWING NORTH END AND EAST SIDE

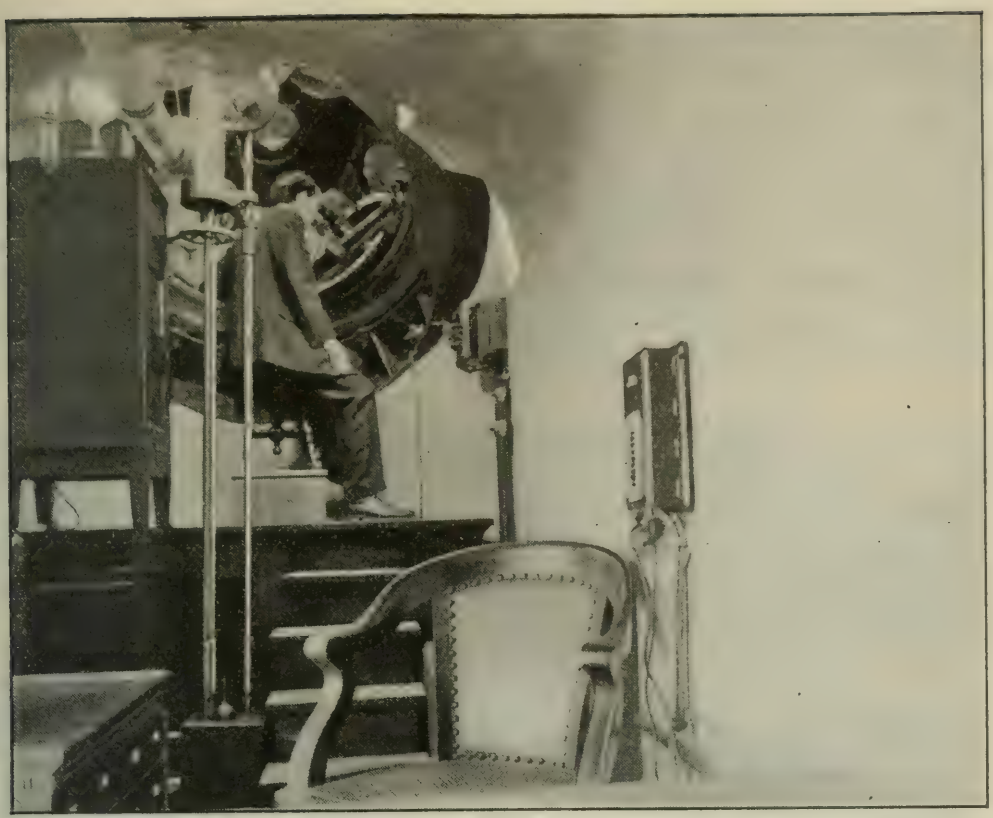


FIG. E INTERIOR OF OBSERVING ROOM

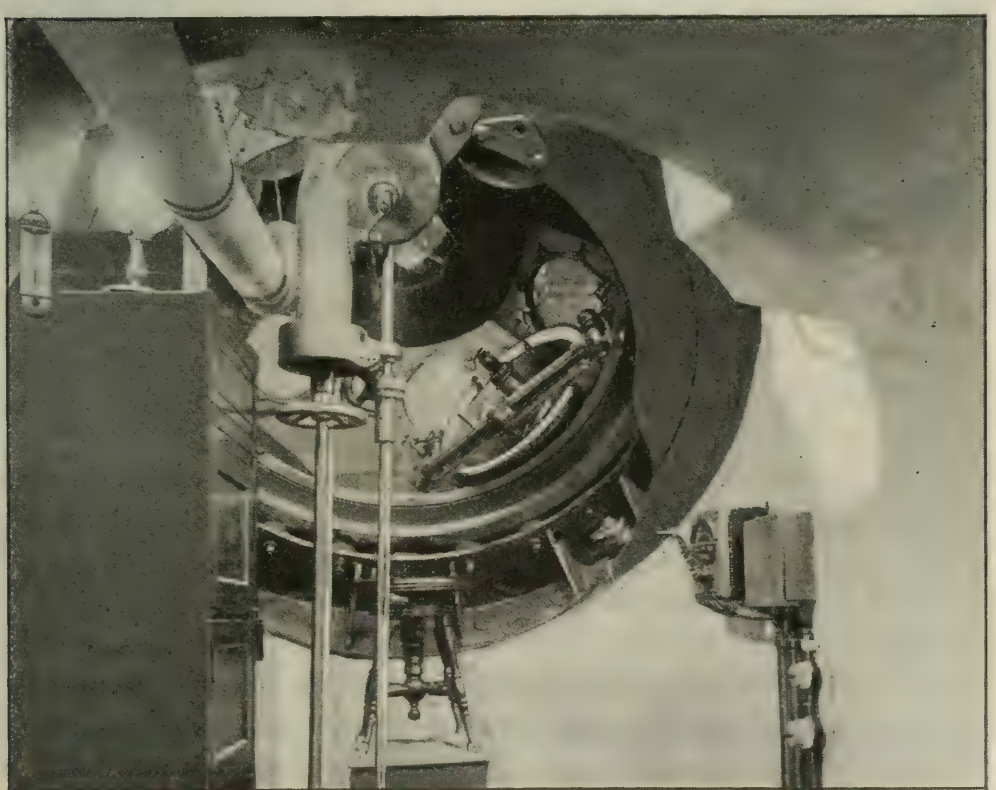


FIG. F ENLARGED VIEW OF INTERIOR OF DOME

is changed, but instead of using a large mirror near the objective, a prism is used nearer the eyepiece. The area of the bundle of the cone of rays at the prisms is approximately $\frac{1}{16}$ of the area of the object glass. By avoiding the use of large mirrors there is, in addition to the optical gain, an improvement in the instrument as a machine by the elimination of all of the mechanical difficulties involved in mounting and controlling the shape and position of these large mirrors.

14 The means employed to shelter the observer and provide suitable mounting and control for this telescope may be briefly described

FIG. G TURRET EQUATORIAL OBSERVATORY AND RESIDENCE TO WHICH IT IS CONNECTED BY UNDERGROUND PASSAGEWAY

as a building having a turret-like dome, this dome being mounted to rotate on an axis parallel to the axis of the earth.

15 In approaching this problem it is perhaps well to assume that the distances between the instrument and the celestial objects are so great as to be practically infinite; hence, in saying that the turret is mounted on an axis parallel to the earth's axis, it will be just as well for our present purpose to consider it located at the axis of the earth. Furthermore, in considering the joint between the telescope and the turret, it will be more readily understood if we regard the pivotal connection of the telescope tube to the dome as located directly in the middle of the turret instead of at one side.

16 Regarding the mechanical precision of the present instrument, it is very natural for us to question the reliability of a turret in serving the place of an arbor for the main polar axis. This axis not only provides a control of the motion of the telescope as it follows a star in offsetting the earth's motion, but it also must combine with it some means by which the exact hour position or right ascension of the object may be known. These two things make it necessary to provide the turret with a perfectly planed surface on its under side and a truly circular track. It also requires in the building a stable mounting for rolls on which the turret rests.

17 There are two sets of rolls. One set on which the flat face of the turret rests must keep the axis of the turret parallel with the axis of the earth; the circular part of the turret bears on the other set of rolls. The office of the second set is to hold the axis of the turret in a fixed position relative to the building to facilitate convenience in measurement of the angular position of the telescope as to hour position or right ascension.

18 If we can assure ourselves of the reliability of the turret as a means of axial control and a means for measuring the angular position of the tube around this axis, the author thinks we should be ready to accept the present scheme on account of its making possible the comfortable housing of the observer.

19 In mechanism for obtaining the greatest precision of axial control of a rotating object, machine designers invariably prefer an arbor of relatively small diameter mounted in two bearings, with the distance between the bearings of at least half a dozen diameters. Small diameters are preferable, too, because they give the most reliable center control around which to measure angular position.

20 In the present case as in nearly every problem of machine design, there are, however, other elements to be considered. The best solution is to determine what compromises to make. Even the almost ideal mounting of the standard telescope, such as exemplified in the great Yerkes refractor, an instrument which undoubtedly measures the highest attainment along these lines, seems handicapped by a delicacy of poise which is not the most favorable for stability of control.

21 It is entirely beyond and outside the object of this paper to discuss these various instruments, but it seems necessary, in order to set forth the object of the present instrument, to call attention to those that have preceded it, and in doing so to set forth the apparent advantages and disadvantages as they appear to a novice, giving a

view which, while it may not be correct, has at least the advantage of being a fresh one.

22 The refracting telescope having standard equatorial mounting is not only the best, optically considered, but it is undoubtedly superior to all others when made in the smaller sizes and for use in comfortable temperature. The mechanical difficulties, however, increase very fast with the size of the telescope, and there are, of course, many latitudes and altitudes at which these instruments are used where the observer must be exposed to very cold weather.

23 The mechanical handicaps of the larger telescopes are due to the overhanging tube and counterweights. Not even the highest excellence in workmanship in making these machines nor the skill in their use seems to offset this instability. To the layman, at least, it would appear that a breath of air or a change of adjustment would be sufficient to cause a quiver.

24 Referring now to the use of large mirrors, such as are used in the equatorial coudé and others, we have already mentioned the mechanical difficulties encountered in controlling the position of these mirrors without distortion. In the tower telescopes for solar work exceedingly thick mirrors have been used, and in the other telescopes devices have been employed by which the mirrors have been equally supported by many contact points; but notwithstanding this, no scheme seems fully to prevent the bending of the mirror and distortion of the image. The problem is especially difficult because these mirrors must be held in so many different positions. It would be a comparatively simple matter if there were no change of position. Furthermore, there is a temperature disturbance which is greater in a mirror than in a refractor. For this reason it is necessary to maintain an even temperature through the mirror.

25 In addition to the distortion of the image by lack of mechanical control of the mirror, there is the serious objection of absorption by even the most perfect reflectors. This absorption reduces the total light that reaches the eyepiece, and since the object in using telescopes of larger light-gathering power is to get more light, the use of a mirror is equivalent to a reduction of the diameter of the objective.

26 Furthermore, this cannot be offset by mere increase in size, for the loss in definition due to atmospheric disturbance increases with the diameter of the telescope, resulting in a net loss in definition. Therefore the price paid for comfort is not only the amount which must be expended for the large object glasses and still larger reflectors to go with them, but also an actual loss in definition. This loss

of definition, of course, is greatest in the low altitudes and least in the rarefied air of the best mountain top observatory sites.

27 In all equatorial telescopes provision must be made for changing the angular direction of the telescope to and from the pole; in other words, north and south. The axis on which the telescope turns for this position of declination must stand, of course, at right angles to the polar axis.

28 In the standard equatorial the arbor which furnishes the polar axis is provided with an opening transverse to it, which serves as a bearing for the arbor that is affixed to the side of the telescope tube and it is on the precision of this axis that the instrument depends for its true position in declination. It is undoubtedly due to the attachment of this secondary axis to the primary axis that this standard equatorial has the appearance of instability. It is, however, nicely counterbalanced, and the great precision of control is due to the excellence of workmanship and manipulation of a very high order. Nevertheless, it would seem desirable to provide some suitable control for the tube itself on this as well as on the primary axis.

29 In the new instrument the telescope is not pivoted at the middle, but at a point near the focus, and it is at this point that the prism is introduced and the rays are delivered into the turret through the hollow declination axis.

30 The scheme of counterweight resembles in some respects the old-fashioned well sweep. This counterweight is fulcrumed at the opposite side of the turret and reaches over to a point near the head of the telescope, supporting it without adding the weight to the telescope head, thus relieving a part of the weight of the declination axis and also all torsional strain.

31 This well sweep acts also as a brace in one direction. Its duty changes from that of a counterweight to one wholly of a brace. For instance, when the turret has been turned so that the declination axis is at the lowest part, and the tube has been pointed directly to the pole, then the counterweight is inoperative as a counterweight for the tube, but the arm becomes a brace. Therefore this scheme changes from one that is wholly a counterpoise to one that holds the telescope in position by a diagonal brace, and, of course, there are positions in which there is an equal service of each in the change from one extreme to the other. The brace always operates to stiffen the instrument against the action of the wind in certain positions and the exact form may be varied to get the best results.

The one shown in the illustrations was made to use available material.

32 The means for controlling the declination position of the telescope is a wormwheel in which two worms engage. All of this is clearly shown in the drawings Figs. 17 to 20.

33 It will be seen, then, that the observer sits inside the dome and receives the light coming from any celestial object above the horizon, and although it is necessary for him to change the position of the chair about 1 ft. east and west and 3 or 4 in. vertically, such a change does not constitute any serious inconvenience.

34 The beam of light comes into the observatory in a horizontal position when the telescope is pointed along the meridian, and it changes from the horizontal position to one in which the observer looks down towards the north at an angle of about 45 deg. at this latitude. The control of the dome, both for its quick motion for changing from one position to another, and also for providing movement to offset the rotation of the earth, is all effected by levers within convenient reach of the observer.

35 The rotation of the dome, instead of being effected by a wormwheel or gearing directly connected to the dome, is through the means used for driving four of the supporting rolls. These rolls take bearing on the circular track. They are hardened and ground to the proper diameter to get the desired relation of speeds. All of the other rolls are merely idlers for maintaining the fixity of axis of dome, and all rolls are mounted in ball bearings to reduce the total power required in turning the dome. Ball bearings keep the resistance more nearly uniform than could be maintained with plain bearings.

36 The motor which furnishes the power was originally located within the building, but on account of its noise it was placed in a separate box outside of the building.

37 As shown in Fig. G this observatory is located a short distance from the observer's residence. It is connected by an underground passageway which not only serves as a shelter in going to and from the observatory, but also as a means for carrying telephone and electric wires, hot and cold water, and the hot water for heating.

38 It is located on the brow of a terrace on which the residence stands, at a level a trifle lower, so that the tunnel rises in going from the lower room in the observatory to the basement of the residence, about 9 ft. in the 240 ft. of length. This difference in level serves as an aid in ventilation, although it makes necessary the use of a small centrifugal pump to induce the hot water to circulate in the observatory.

39 The location of the observatory on the brow of a hill or terrace, makes it possible to get a fresh air inlet from the lower part of the lower room by running an air duct out through the side of the bank to the open air. In cold weather the air rushes through this air duct into the observatory, through the ventilators in the top of the door into the tunnel, and thence out of the tunnel either to the open air at the head of the tunnel where it joins the residence, or through the house. In summer, when it is hot outside, the current of air flows in the opposite direction. At such a time the door to the residence at the head of the tunnel is kept closed and another door is opened to the outside air. This takes the fresh air from the outside, which travels down through the tunnel by gravity, giving up some of its heat to the tunnel walls and reaching the lower room of the observatory at a fairly comfortable temperature. Then, by the aid of a small portable fan this air from the lower part of the lower room of the observatory is blown through an air duct into the dome.

40 In this connection, perhaps, it would be well to state that the dome is lagged with wood on the inside and the windows are all double, that is, there is one set of windows in the wooden lagging on the inside, and another in the metal dome outside. This air space is desirable in winter as well as in summer.

41 Regarding the form of the building, this may be almost anything, from a large spherical dome to one having a hip roof. The one feature essential to this scheme is the proper neck for supporting the turret. This neck should be substantially the same as that shown in the present illustrations. The author selected the present form, partly to make it the least conspicuous from his residence, and was perhaps influenced by the earlier form of the scheme in which it was intended to use a reflector instead of a refractor.

42 Closely connected with this subject is, of course, the cost; and this must be based on the way such things are produced. If more instruments of this kind are to be made at different times and in various places, there is no hope of getting them produced at a low figure. There seems to be only one way of insuring the success of an instrument of this kind Every machine builder knows it, and yet the mere mention seems to establish an antagonistic attitude at once in the mind of the average man.

43 It has not been considered ethical to take out patents on scientific instruments, and although the writer has not the courage of a reformer to wage battle against this sentiment, yet when it is such common knowledge that machinery cannot be successfully and eco-

nomically built without the concentration of the energies of a number of men for a given purpose, and since this cannot be accomplished without patent protection, it has been thought best to apply for patents on the new features of the present instrument. Although this will be done without dedicating it outright to the public, it is needless to say that no barrier will be allowed to prevent these instruments being built by any one until some business arrangement is made for the exclusive manufacture of some one size by some builder, and even then, others may be permitted to build other sizes; but all such permission will be given only by letter and not in a broadcast way that would in any way handicap the main purpose of making these machines available at a low cost to anyone who may desire them. It goes without saying that with patents there is always the thought of the exclusive right and profit of the patentee and manufacturer, but regarding this point it is well-known that the low cost of such things, as well as the best workmanship and results, can be obtained only by concentrating all of the work in one plant.

44 The writer does not contemplate manufacturing these instruments. More may be built experimentally, but not for the market. If others wish to build instruments of this kind, permission will doubtless be freely given, but with certain restrictions; but all this must be arranged in each case by correspondence.

45 If patent is granted no charge of any kind will be made for license to build until some arrangement has been made for manufacture on an efficient scale, and then only such restrictions as in the opinion of the patentee will be for the best interests of the science.

46 In closing, the writer begs to call attention to the fact that the great advancement in the world's knowledge, due to the work of the astronomers, has been carried on by the men who have braved the mosquitoes in summer and observed long hours in the most unfavorable temperatures. These men have been recruited from those who have taken up the work as an avocation and for every man of this number there have undoubtedly been ten or a hundred who have grown faint-hearted at the mere thought of the exposure incident to observing and at the high cost of the large instruments. With means of this kind made generally available it is thought that there will be more men in this work, and that from the greater number perhaps even greater work may be accomplished in the future.

DATA REGARDING THE TURRET EQUATORIAL

Object glass, 10 in. in diameter, and 150 in. in focal length.

Inside dimensions of building, 6 ft. by 18 ft.

Outside diameter of dome, 7 ft., 4 in.

Weight of dome casting, about 1½ tons.

Total weight of dome with counterweight, tube, etc., about 2 tons.

Weight of ring-casting on which the dome is mounted, 1½ tons.

The optical parts were furnished by The John A. Brashear Co., Ltd. All of the other work was home-made, except the two large castings which were cast and turned in Fitchburg, Mass.

The hour circle consists of a flat ring having approximate dimensions of 52 in. outside, 48 in. inside, and ¾ in. thickness.

The inner diameter is graduated down to divisions of 1 minute. These divisions are about 0.1 in. apart. Vernier edges were provided for division of the minutes into seconds, but thus far a crude substitute has been used in preference. It consists of a rotating dial driven by gearing which also turns the dome. Its connection is such that the dial makes 1 r.p.m. With the 60 graduations on dial, the division of time into seconds is very readable.

In use the hour circle may be set to the even minute of position, leaving the plus or minus of seconds to be allowed for at the zenith point by the time piece.

The declination wormwheel is 24 in. in diameter, and is provided with two worms located on opposite sides. These are rotatively connected by means of a cross shaft and spiral gears.

In order to prevent conflict of action an eccentric bearing is provided for accurately gaging the depth of engagement of each worm. A separate means for turning each a very slight amount is also provided.

Graduations for minutes of degrees are carried by hubs on worm shafts.

The proximity of the two eyepieces led to the disuse of the finder, but if it is needed in future work, there are a variety of schemes for overcoming this difficulty.

Attention should be called to the rigidity of the eyepiece and the opportunity afforded for attaching photographic and spectroscopic apparatus. This feature is to be found in other telescopes designed for comfort, but as this instrument seems to partake more of the good seeing qualities of the standard telescope, it does not seem out of place to call attention to this advantage it has over the standard in which the delicacy of the poise is disturbed by added weight and adjustment of counterbalances.

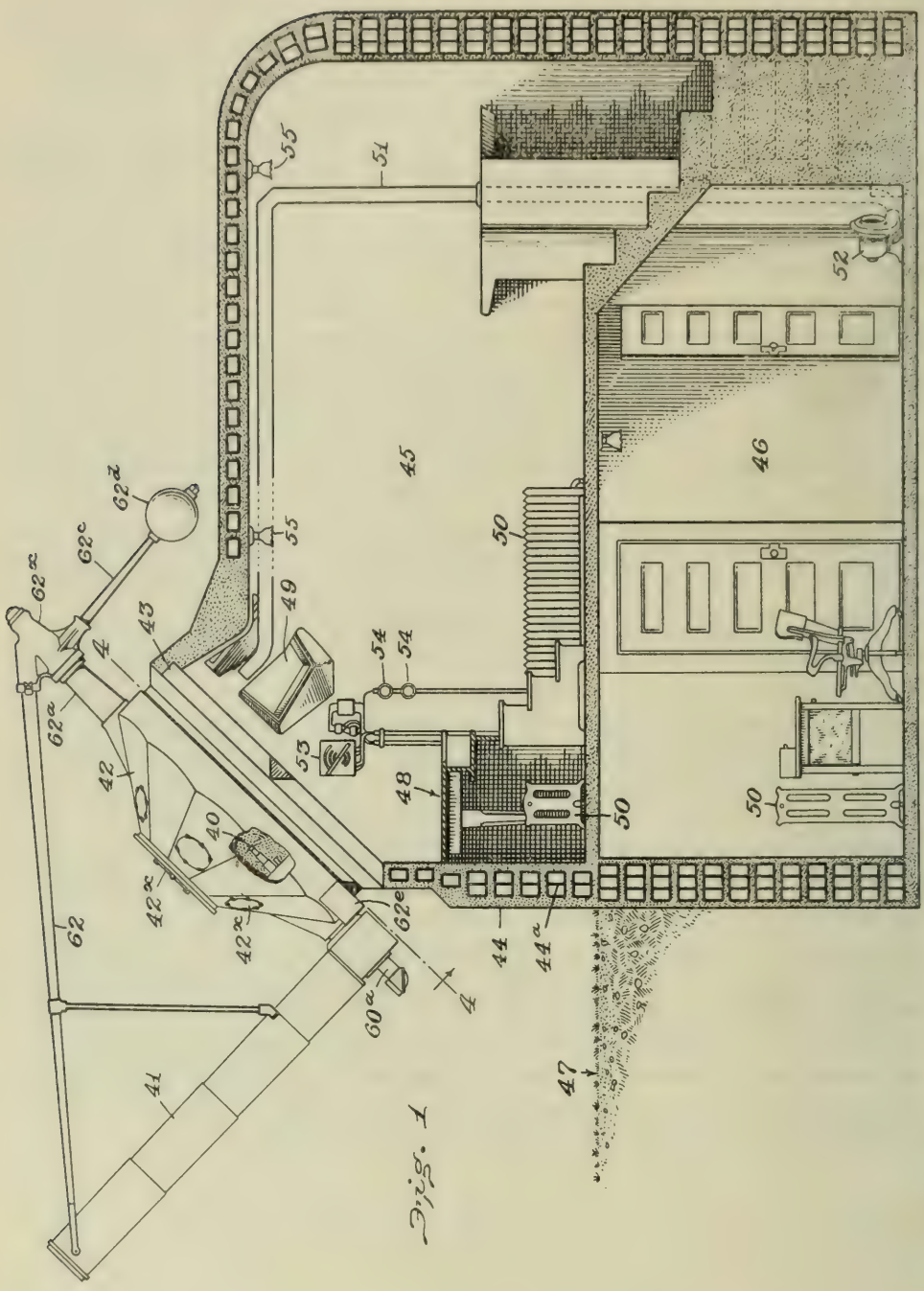


Fig. 1 Side Elevation of the Telescope and Dome and a Longitudinal Vertical Section of the Building which forms the House for the Observer

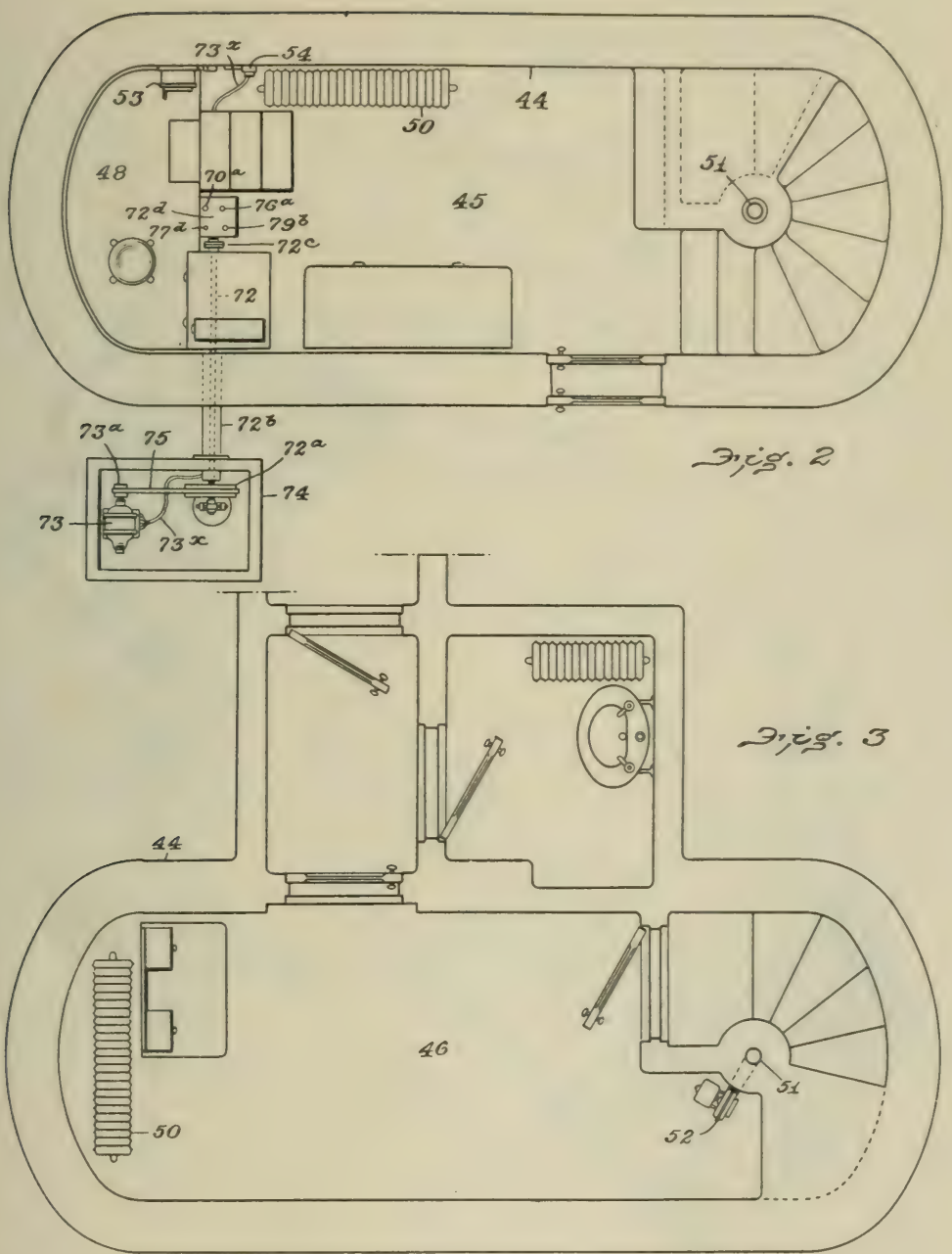


Fig. 2

Fig. 3

Fig. 2 Floor Plan of the Upper Compartment of Observing Room and the Box outside the Building in which the Motor is located

Fig. 3 Floor Plan of Lower Compartment and Subway Entrance

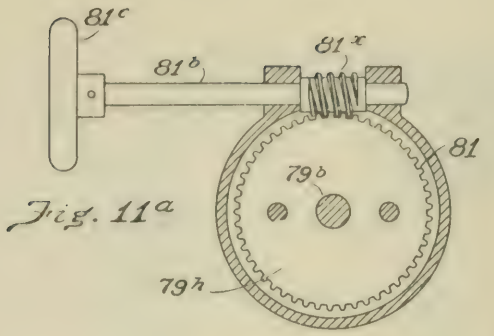


Fig. 11a

Fig. 3a

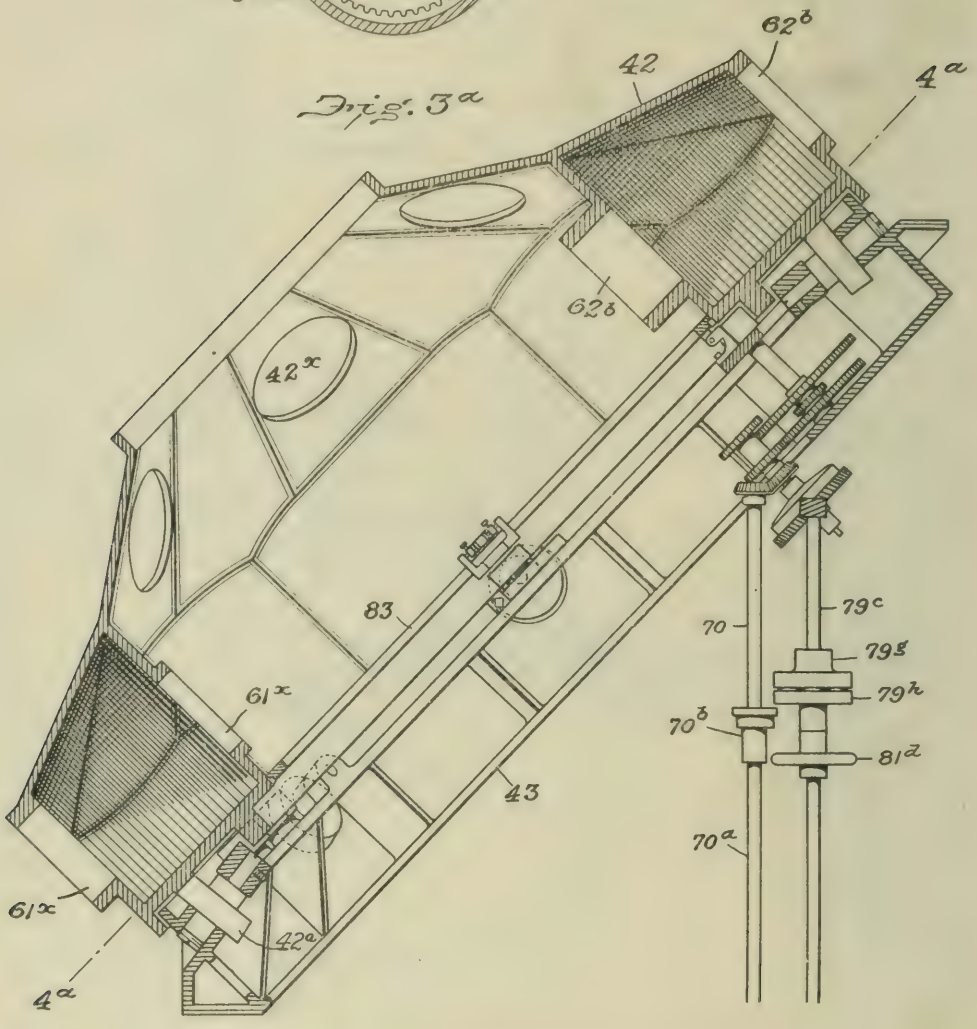


Fig. 3a Longitudinal Vertical Section through the Turret

Fig. 11a Original Form of Differential Motion for manually varying the Advance of Dome as it is turned by Motor to follow a Star

This has been superseded by Fig. 11, but may be reinstated again.

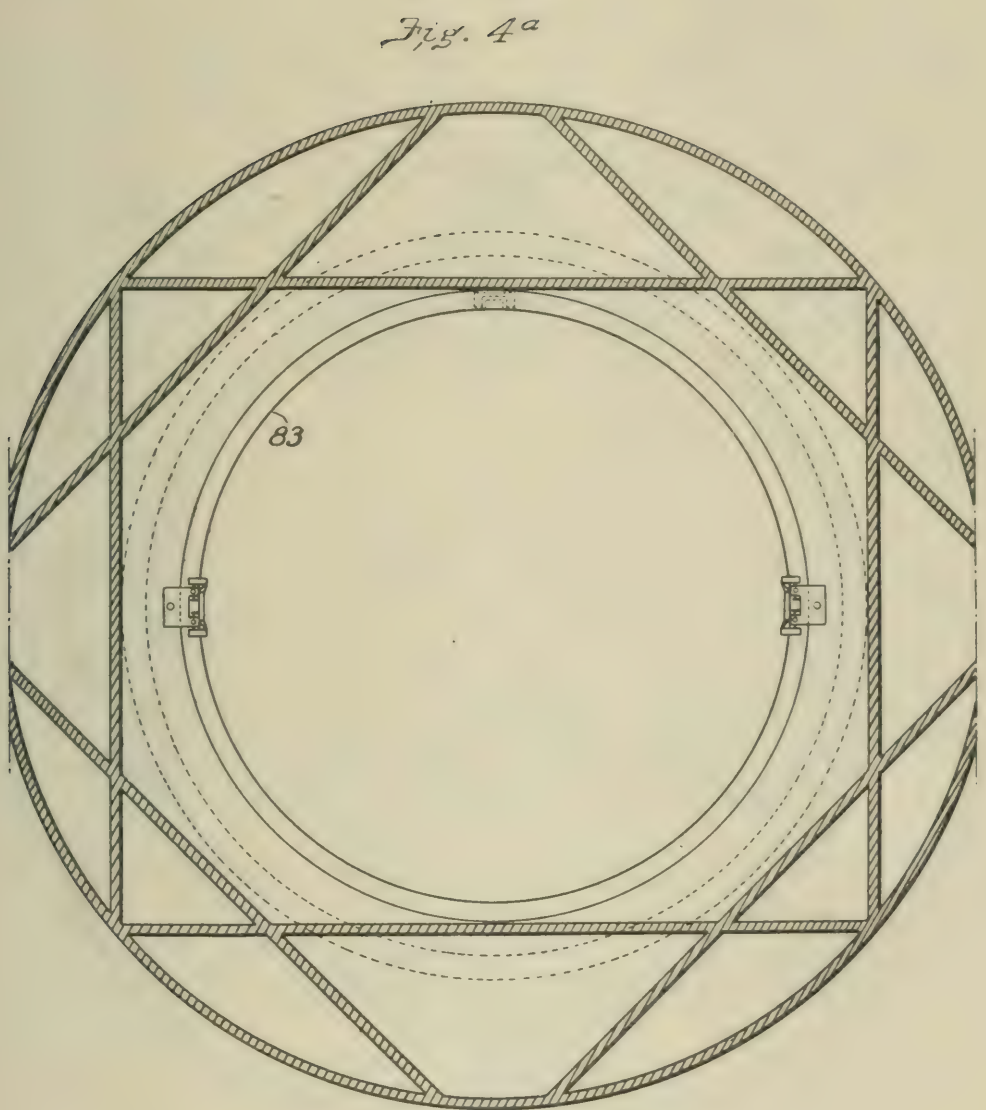


FIG. 4a CROSS-SECTION THROUGH TURRET ON THE PLANE INDICATED BY LINE 4a4a IN FIG. 3a It clearly illustrates the scheme of ribbing which was chosen to resist warping or change of plane of under side of turret. It also shows its stability to resist any tendency to lose its true circular form of track, 42f in Fig. 4. The entire precision depends on the reliability of this plane and circular face of turret. This may be considered the most essential element in the whole structure.

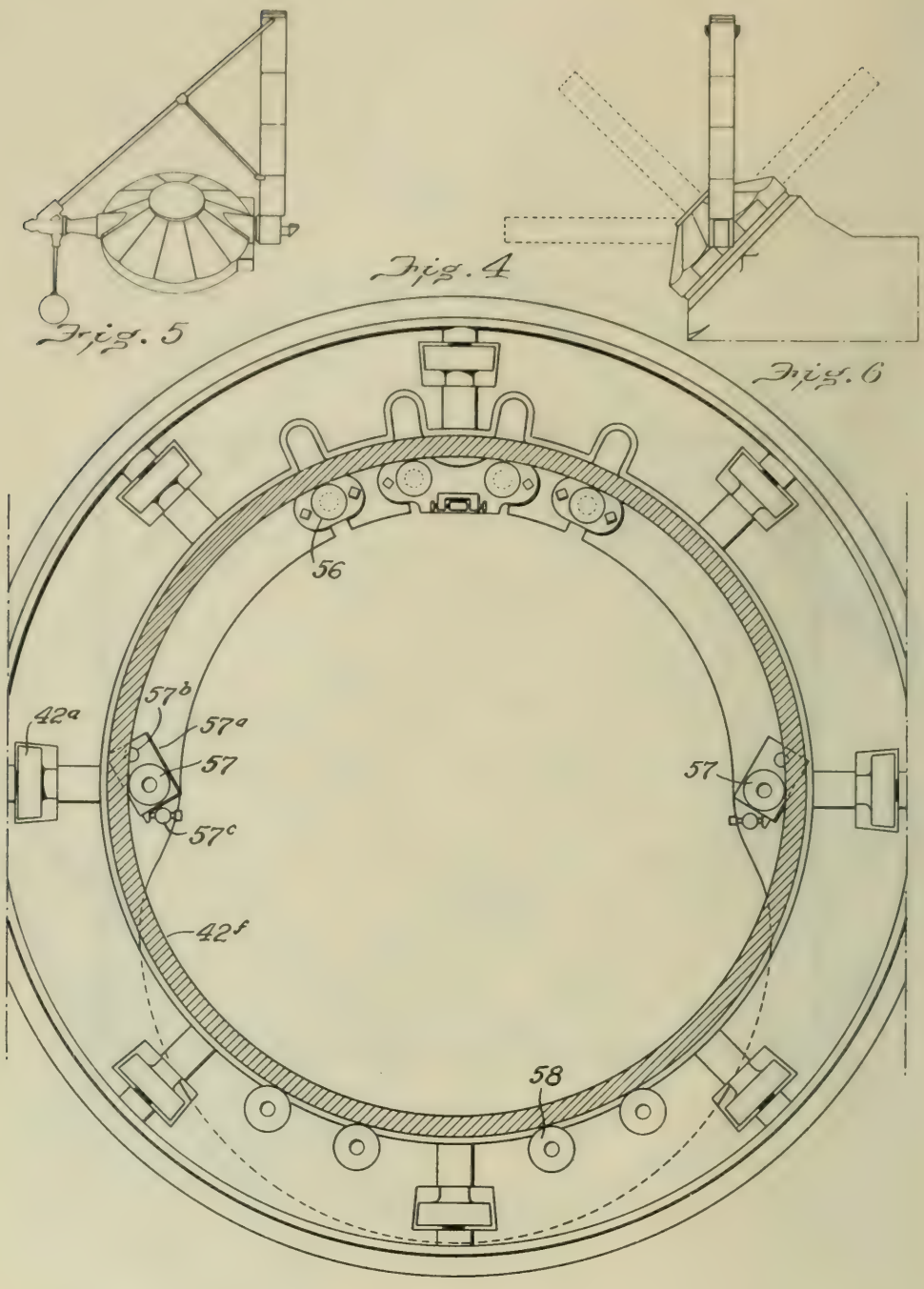


FIG. 4 SECTION THROUGH THE TURRET INDICATED BY LINE 4—4, FIG. 1

The eight conical rolls 42*a* furnish a support to plane the face of the turret. Their office is to keep the axis of the turret truly parallel with the axis of the earth; that is, they prevent the turret from tilting. The four rolls at the top, marked 56, are the driver rolls. The circular track of turret is shown in section 42*f*. The turret hangs on these driving rolls held in position laterally by side rolls 57.

The four rolls marked 58 at the bottom of the figure are to relieve the burden on driving rolls 56 when the instrument is not in use.

FIGS. 5 AND 6 NORTH AND WEST ELEVATIONS OF THE TELESCOPE AND DOME SHOWING TELESCOPE POINTED TO ZENITH; ALSO BY DOTTED LINES, VARIOUS OTHER POSITIONS OF THE TELESCOPE RELATIVE TO THE DOME

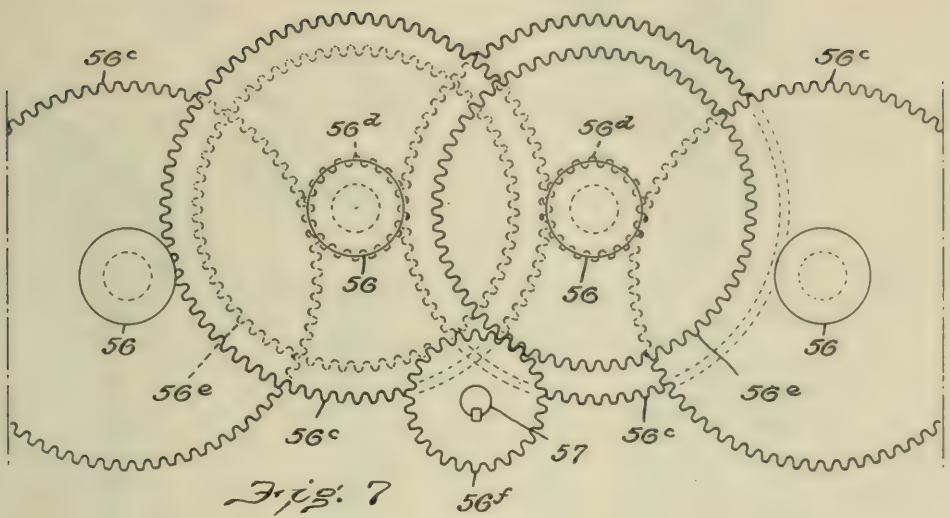


Fig. 7

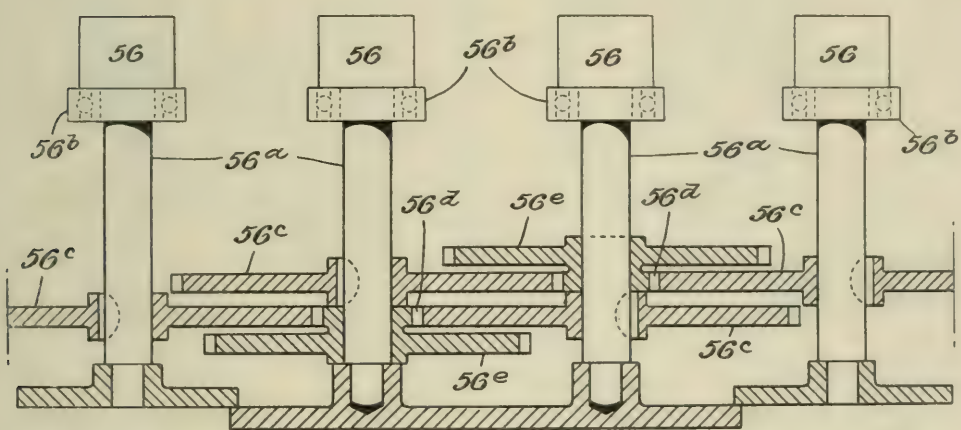


Fig. 8

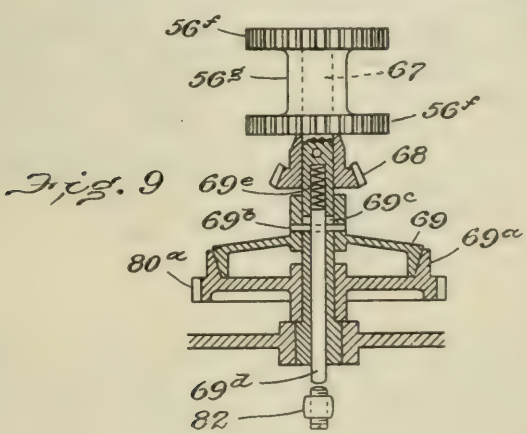


Fig. 9

FIGS. 7 AND 8 PLAN AND ELEVATION OF DRIVING ROLLS AND THEIR GEARS
FIG. 9 CLUTCH MECHANISM FOR THE SLOW MOTION OF DOME (SEE FIG. 10)

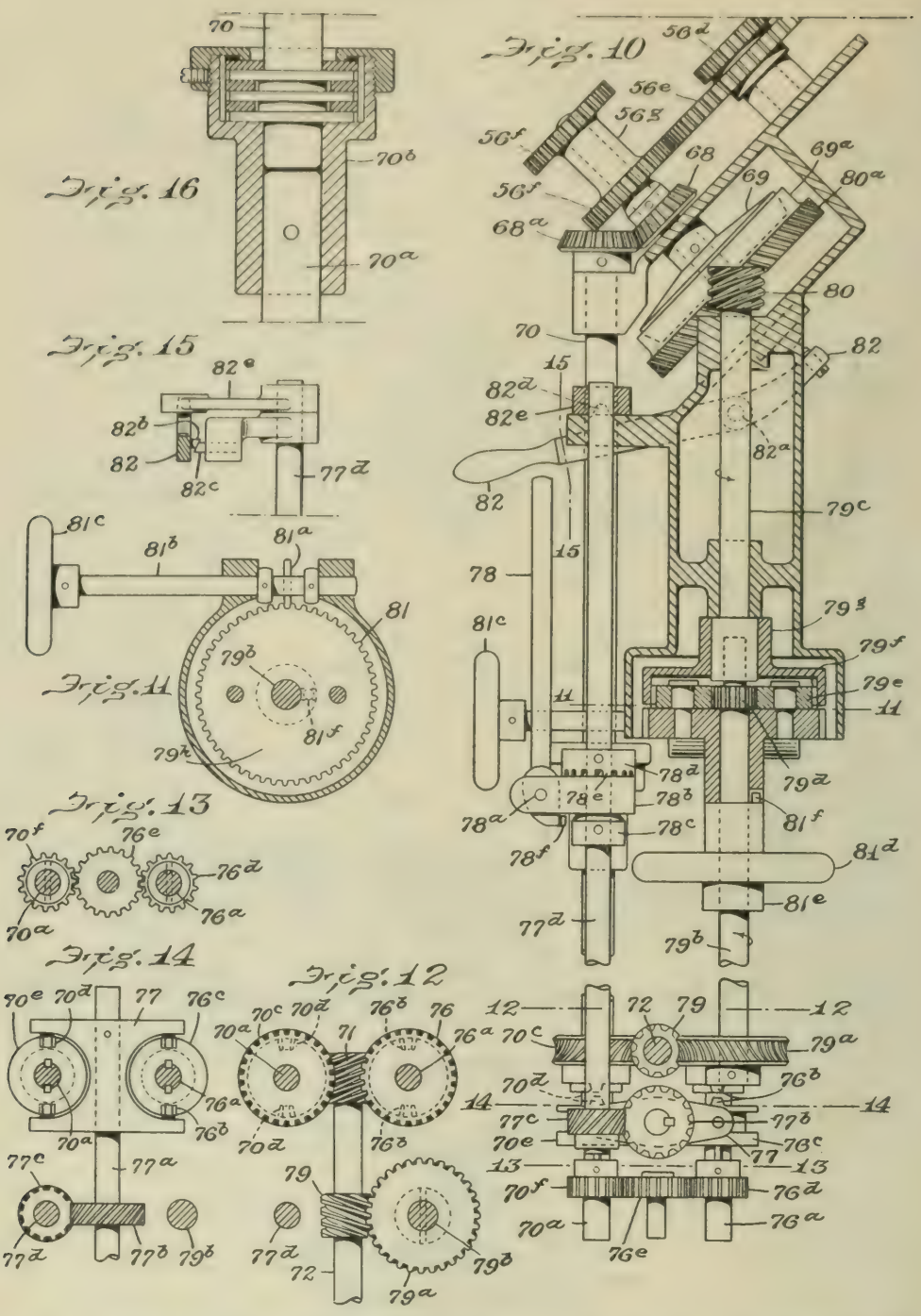


FIG. 10 DIAGRAM SHOWING TRANSMISSION GEARING AND CLUTCHES THAT GIVE THE MOTIONS TO THE TURRET. IT EXTENDS FROM THE TOP TO THE BOTTOM OF THE PAGE

The fast motion forward and back is under control of lever 78 and the slow motion is controlled by lever 82. These levers interlock to prevent conflicting engagement. The slow-motion clutch is shown as a friction clutch at 69 and 69*a*, and the fast motion clutches are of the positive, or toothed-clutch type and are shown rather obscurely at 70*d* and 76*b*. These gears with their clutches are located in a gear box on the floor of the observing room (Fig. 2). The shafts 70 and 70*a* which transmit the motion from this box to the upper group of gearing are unfortunately obscured in this view by the lever control shaft, 77*d*. They may be more clearly seen in Fig. 3a.

FIG. 16 SLIP BOX OR SAFETY COUPLING BETWEEN THE TWO SHAFTS 70 AND 70*a*

It is wholly unnecessary, for the driving rolls slip when any undue resistance is encountered.

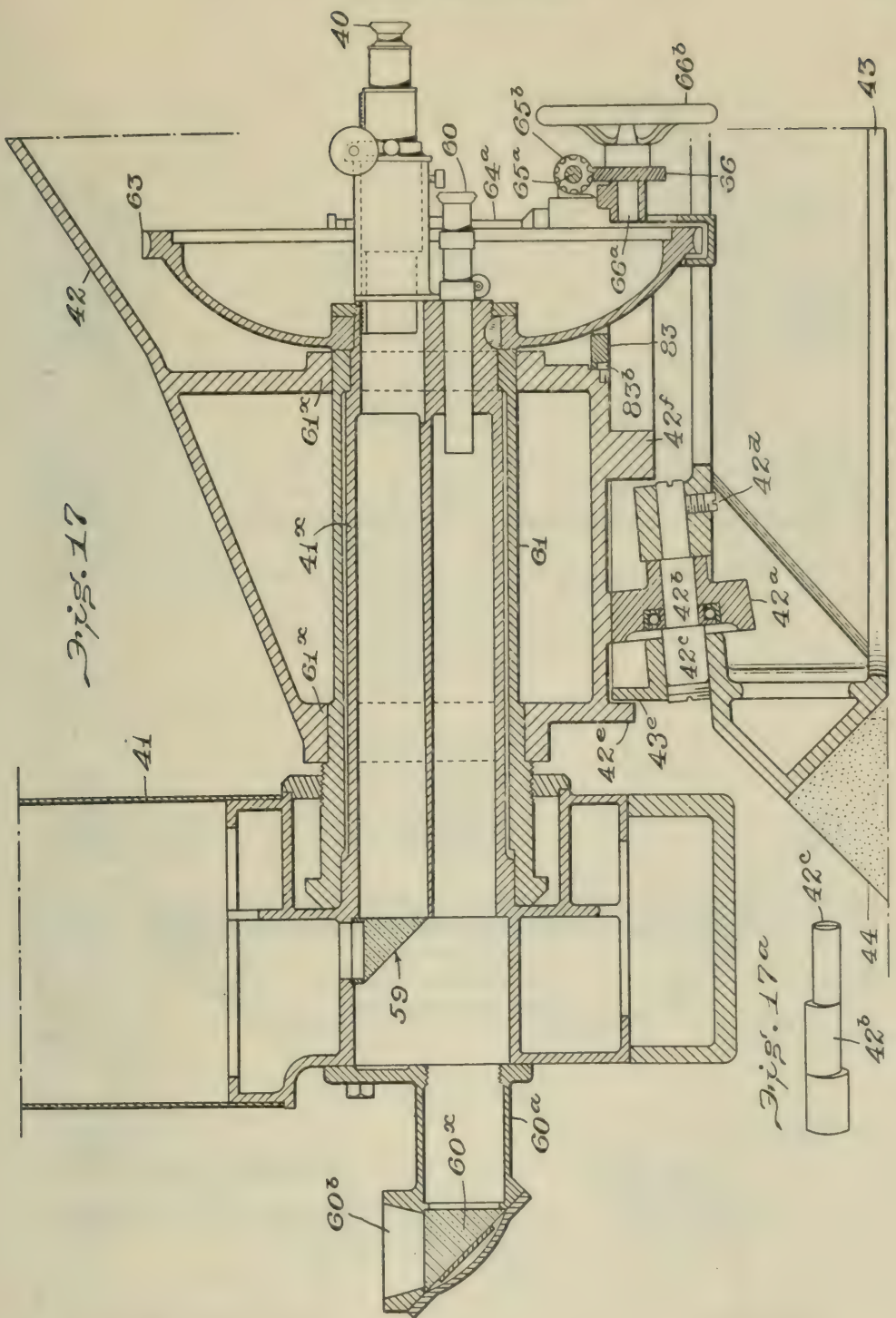
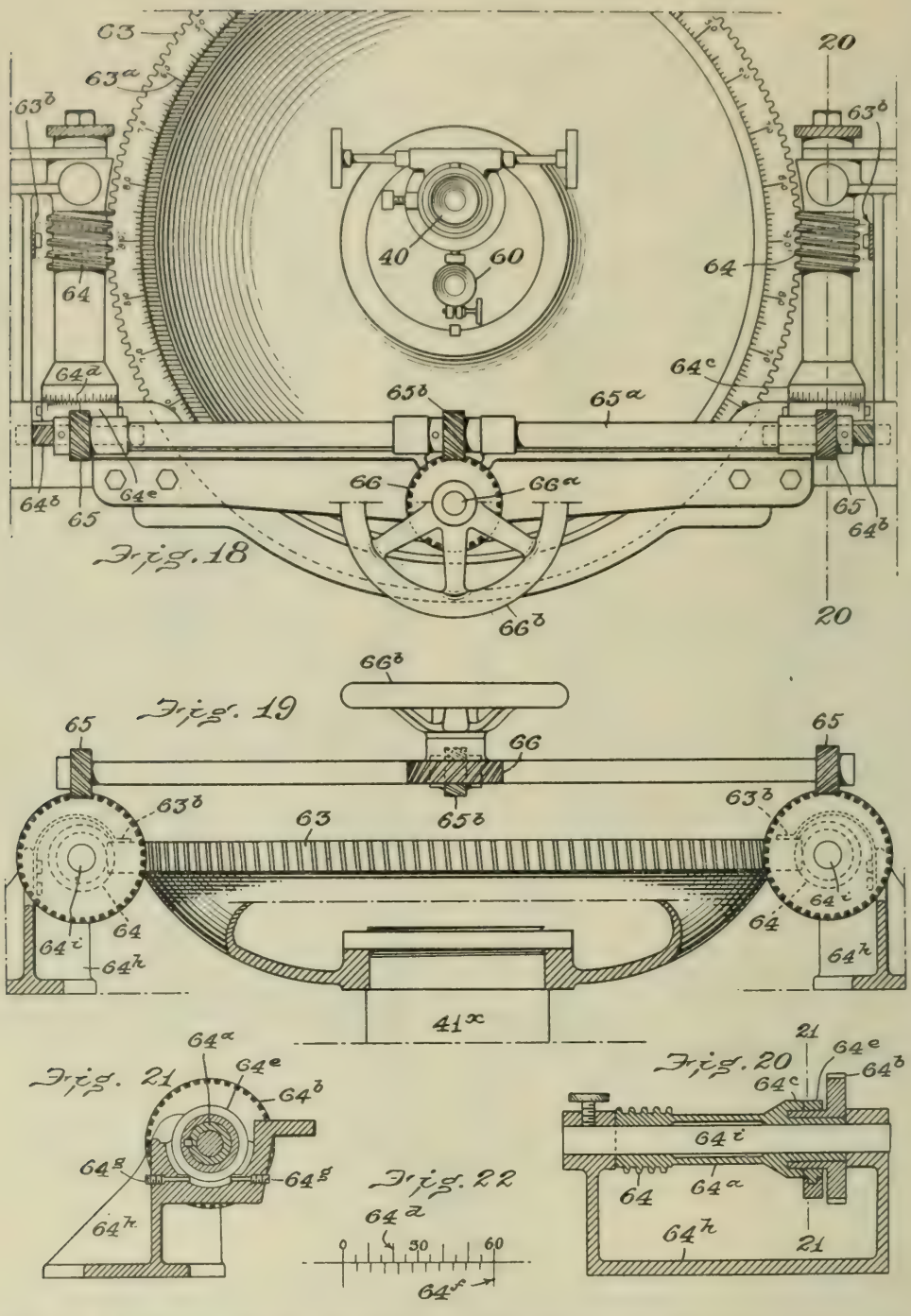


Fig. 17 Sectional Portion of the Turret through the Declination Axis, showing also Section of the Lower Part of Foundation Ring with one of the Conical Supporting Rolls in Section

Fig. 17a

The part of the foundation ring shown is the same as the lowest part shown in Fig. 3a. In Fig. 17 the end of the telescope tube is represented by 41. It is attached to the head declination arbor 41x. This arbor turns in bushing 61 which is forced into turret seats at 61x and 61z.

The cone-shaped beam of light from the object glass comes down through tube 41, passes through prism 59 which changes its direction 90 deg., delivering it to the focal point near ocular 40. The light for the finder enters through opening 60b directly into prism 60x, thence through an object glass (not shown) which is located in the inner opening of 60a. From this object glass it converges to focal point near the finder eyepiece 60.



FIGS. 18 TO 22. VARIOUS VIEWS OF DECLINATION WORMWHEEL AND ITS CONTROL BY DOUBLE WORMS

The form of the wheel and its connection to the arbor is not ideal, but may be very much improved in any later work.

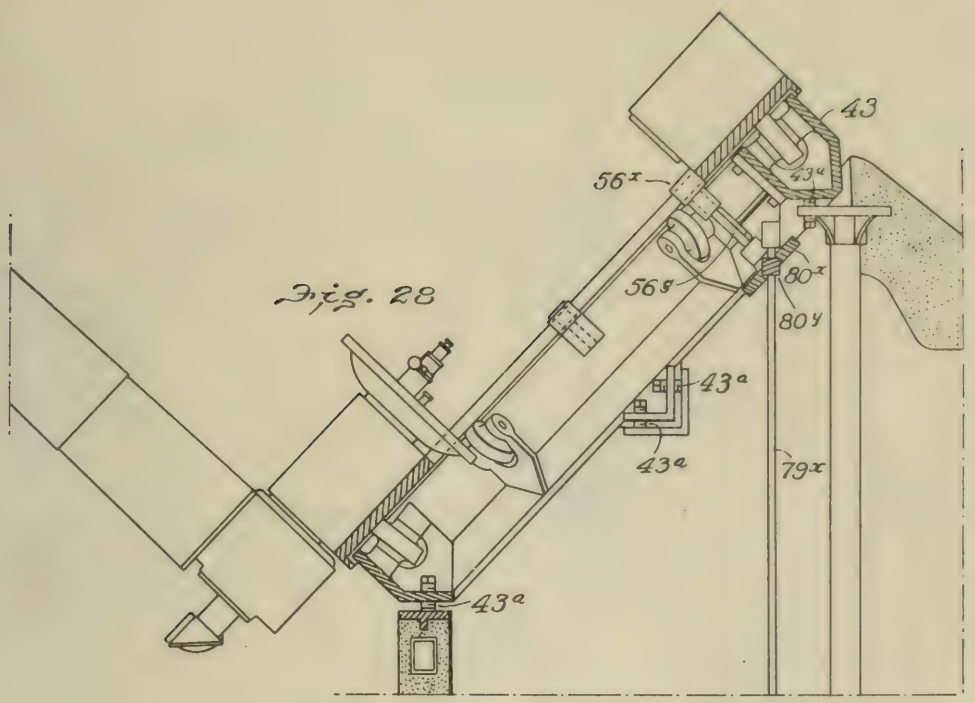


Fig. 28

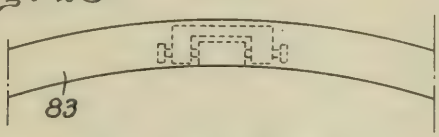


Fig. 23

Fig. 24

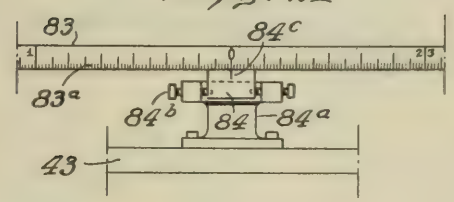


Fig. 25

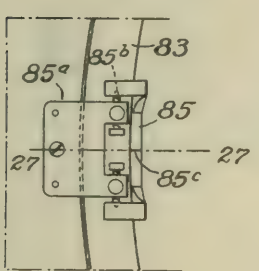


Fig. 26

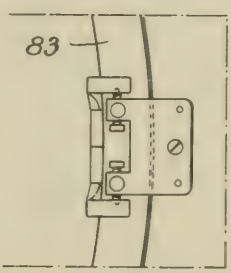


Fig. 27

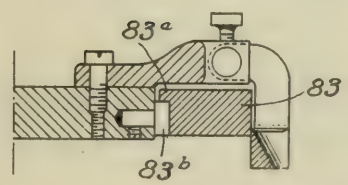
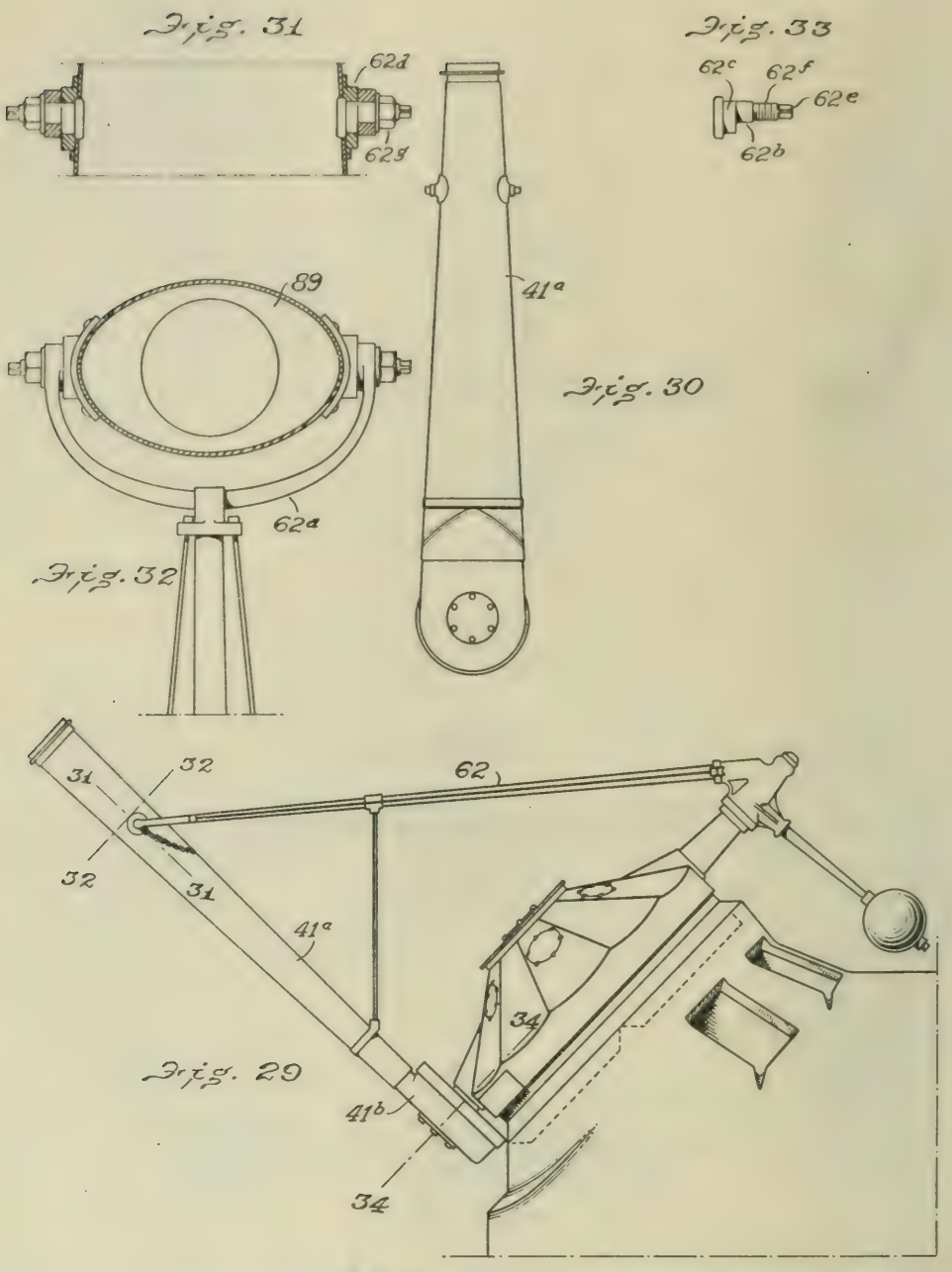


FIG. 28 A SCHEMATIC ARRANGEMENT OF THE ESSENTIAL ELEMENTS IN THE MOUNTING OF THIS TELESCOPE, NAMELY, THE FLAT CIRCULAR RING MOUNTED ON CONICAL AND CYLINDRICAL ROLLS, MEANS FOR ADJUSTING FOUNDATION RING AND FOR DRIVING ROLLS

Smaller turrets might be made without use of supporting rolls and by going back to the worm drive instead of the roller drive. A great variety of circular hollow bearings might be used for smaller turrets, but for one of the present size a single true plane and a single circular track with rolls seem to be the best.

FIGS. 23 TO 27 VARIOUS VIEWS OF VERNIERS FOR HOUR CIRCLE



FIGS. 29 TO 33 DIAGRAMS SHOWING ONE OF THE MANY POSSIBLE MODIFICATIONS OF DECLINATION CONTROL

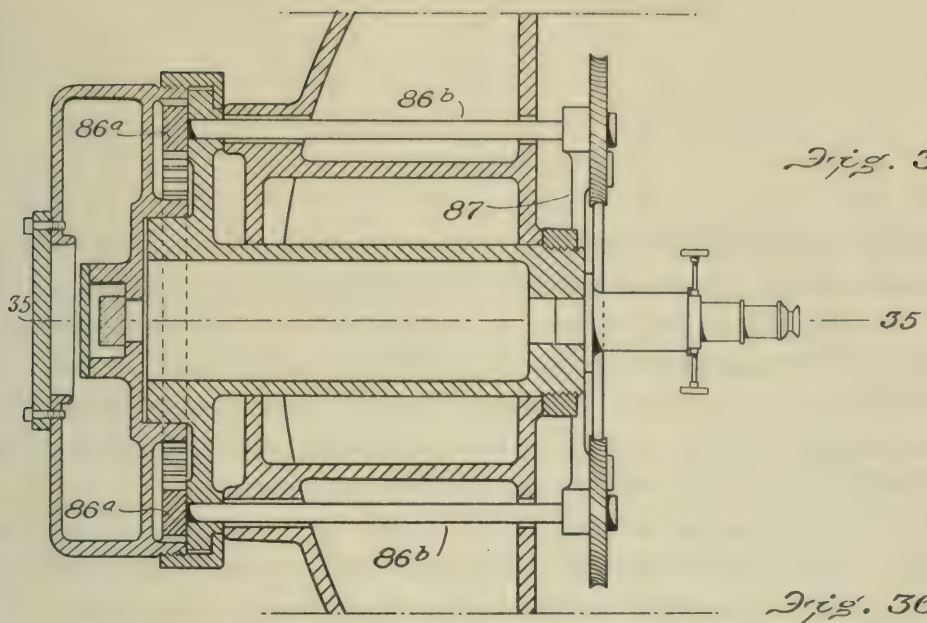


Fig. 34

Fig. 36

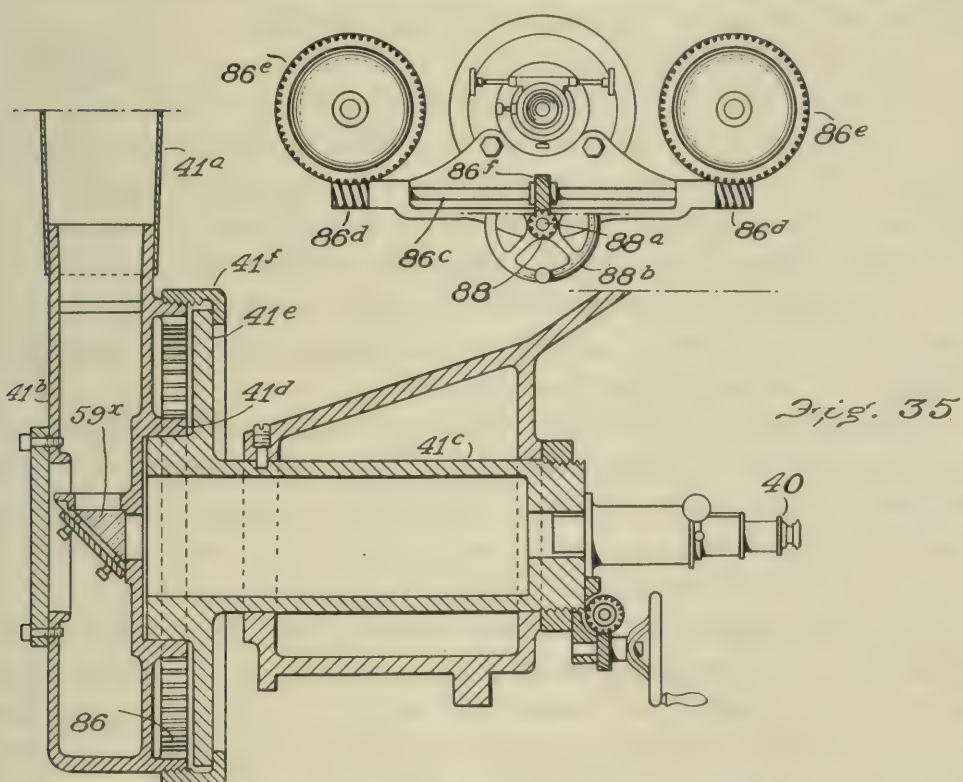


Fig. 35

FIGS. 34 TO 36 DETAILS OF SCHEME SHOWN IN FIGS. 29 TO 33

DISCUSSION

Worcester R. Warner. It is an interesting fact that some twenty years ago the author of this paper, having given his attention to the study of the turret lathe after its known virtues had been threshed out by many eminent engineers for a generation, designed and developed the machine on new lines which proved so successful that after he had made his fortune and his patents had expired, several leading manufacturers in this country and in Europe copied it.

The turret principle has been successfully applied to our warships, and in many other mechanical devices, but who would have guessed that it could be adapted to the design of an equatorial telescope for the observation and study of the heavenly bodies?

Mr. Hartness did it, and his design is so unique, so sensible and so admirable, that I am reminded of the Boston gentleman who said, after reading for the first time a volume of Shakespeare's plays, "I don't believe there are fifteen men in Boston who could have written that book." I believe there are not fifteen men or two men in this Society who would have thought out a method of applying the turret principle to the equatorial telescope in such manner as to allow the astronomer to remain in his warm room and observe at will any star in the visible heavens.

An interesting coincidence is that just three hundred years after Gallileo pointed his first telescope to the heavens Mr. Hartness brought out his turret equatorial telescope of such size, and the Brashear optical parts of such perfection, that it will show stars that Kepler and Newton and even Herschel never saw.

It is true that methods have been previously devised which allow the astronomer to work in a comfortable room, but in each case there was great loss of light on account of the large reflecting mirrors employed, while in the Hartness turret telescope the single total reflecting prism is so near the focal plane that it is comparatively small, and therefore the optical features of the instrument retain practically the same high efficiency that is found in the present standard type of equatorial telescope.

In the standard telescope of this size, the polar axis would be about 4 in. in diameter, and 2 or 3 ft. long, its inclination from the level being equal to the latitude of the observatory. With all its attachments it would weigh about 500 lb. The driving clock revolves this light and highly perfected polar axis with its attachments in sidereal time.

In Mr. Hartness's design, there is substituted for the polar axis a casting 7 ft. in diameter, weighing, with its attachments, 4000 lb. Its axis of revolution must be inclined, as in the standard telescope, to equal the latitude of the observatory. In this position it must revolve in sidereal time. To make this movement as easy as possible, it is provided with two sets of rolls.

Any irregularity in the motion of a telescope when, by its driving clock, it follows the apparent motion of the star, is magnified precisely as the image of the star itself. The 10-in. object glass made by Brashear for this telescope will admit a magnification of 1000 diameters. It is fair to infer, therefore, that Mr. Hartness's greatest difficulty will be found in his endeavors to revolve his heavy and necessarily coarse mechanism with sufficient exactness to enable the astronomer to make satisfactory observations, especially the measurements of satellites and double stars.

In the standard type of telescope, the central section of the tube is attached to the declination axis, thus reducing the flexure of the tube to a minimum. By this means the flexure is always constant at equal angles, and can be easily computed and allowed for.

In the turret telescope under consideration, the end of the tube being attached to the declination axis, the flexure of the tube, if unprovided for, would be greatly multiplied. This condition is met by two rods or braces, obviously efficient only when the declination axis and the tube are in the position shown in the drawings.

The emphasis of the author's study seems to have been given to the construction of a large telescope which in the coldest weather can be used while the astronomer is in a comfortable room. I congratulate him on having so successfully solved his interesting problem.

Ambrose Swasey. This paper is a valuable contribution to the papers which have been presented to the Society, and more than that, the instrument itself is a valuable and practical contribution to the instruments used in connection with the science of astronomy. A few years ago we were called upon to provide instruments for an

observatory in one of our larger cities, and when the time came to set up the transit instruments in the room where the temperature must necessarily be adjusted as nearly as possible to that of the outside, it was found that the astronomer had endeavored to be kind to himself and had provided a well-equipped steam heating apparatus for that room. It is needless to say that this apparatus was not used.

If one could spend a few minutes with Professor Barnard and Professor Frost at the Yerkes Observatory on those nights when the thermometer is down to more than 30 deg. below zero, and could see them with their furs on so that they look like Esquimaux, he would feel there was some reason for Mr. Hartness and others giving attention to this question of providing a proper way in which the observer could take a little comfort while carrying on his scientific work.

The French, as indicated in the paper, have perhaps done more than any other people in attempting to solve this problem in the Equatorial Coudé. While that instrument accomplishes its purpose to a certain extent, its mirrors have given any amount of trouble, not only because of the deflection, but because it was impossible to keep them in good condition. In the large refractory telescope with an aperture 50 in. in diameter exhibited in Paris in 1900, the telescope tube remained stationary, and rays from the heavenly bodies were reflected through the object glass to the eye end by means of a reflecting mirror. That was another plan whereby the astronomer might remain in a comfortable place, and in a comfortable position. Its failure was not due entirely to the principle, but because it was so large it was not well carried out. The instrument was a failure from an optical, but not from a mechanical point of view.

Astronomers have had great difficulty owing to the oscillations and vibrations in large refractory telescopes. The revolving dome is an objection which Mr. Hartness does away with and he also eliminates the elevating floor. The dome of the large observatories protects the large glass tubes of the instruments from wind pressure to a certain extent; when supplemented by the auxiliary shutter and fittings, the instrument can be kept in a practically stable and uniform way.

In the large instrument at the Yerkes observatory, about 25 tons are in motion when it is revolving on its axis in sidereal time. In other words, the hand of the dial weighs 25 tons and the clock pro-

pels the instrument so accurately that there is a variation to exceed only a small fraction of the width of the spider line, which is not more than 0.006 in. in diameter.

The paper shows that the telescope is simply another piece of mechanism in which mechanical engineers are interested, and that men who can design and construct other pieces of machinery, are able to construct these higher pieces of mechanism which come under the name of "instruments."

WM. KENT. While Mr. Hartness has been congratulated on his improvements in the construction of the telescope, I wish to congratulate him on the bold, clear and correct statement concerning patent rights. I did not know where the idea arose that it was unethical for a scientific man to take out a patent on a scientific instrument. It is a false notion, because if there is one thing that will actually prevent the manufacture, development and progress of an instrument, it is the fact that there is no patent upon it. If you want a thing put in the hands of the people as much as possible, get a patent on it. I believe that the electromagnetic theories of Joseph Henry did not come into use for some years because he did not take out a patent; if he had procured a patent upon his discoveries, the world would have had the benefit of them many years sooner.

If the inventor of a scientific instrument has not the money with which to take out a patent, he should try to get someone to furnish it. After the invention is patented, the manufacture of the instrument should be turned over to the manufacturer engaged in that line so that the public may receive the benefit of the invention. If a patent is secured and the invention properly protected, a price can be fixed for it which will be reasonable. Do not omit to take out a patent on a scientific device because of the idea that it is unethical.

HENRY HESS. Mr. Hartness and Professor Kent have both taken the position that the promoters of the original patent laws undoubtedly had in mind the encouragement and development of the domestic arts and manufacturers. Probably the present law makers would not pass the patent laws again in anything like their present shape, because they have found as a matter of actual experience that the patent law does not work that way. I do not think there is an inventor in the country who has not a long tale to tell about the things

which are shelved. If you want to have a thing pushed, you must get it into the hands of someone to whom it will be of financial advantage to have the idea patented, which means you will put the matter in the hands of a manufacturer in that particular line. The chances are that if the improvement is a really good one the manufacturer will acquire it, particularly if he can acquire it for the mere cost of the patent, and in many cases he may put it on a shelf so that he may not be compelled, through his competitor, to convert into the scrap heap his previous devices. He will go on manufacturing his existing products as long as he can find a market for his goods at a remunerative price.

F. R. Hutton. The author apologizes for making the polar axis of his telescope of such large diameter and of such short length of shaft, because he thereby violates a principle of machine design. It occurs to me that the point of error is the ball or roller on which the turret is mounted, and not the overall diameter of the ring. In that view there is no transgression of the principles of machine design, in that the accuracy of that ball can be made as close and accurate as the accuracy of any revolving shaft. The dome does not make a complete revolution through 360 deg., so that the mass diameter of the turret is not concerned in the irregularities which are introduced. Such irregularities exist in the raceway and in the ball; and they, it seems to me, can be corrected effectively.

If there is a difficulty in the turret, it will be due to the fact that it stands at an angle, causing it to be heated more at the top than at the bottom through heat rising from the building. The consequence will be that the expansion, due to the unequal heat, will tend to put the turret out of line.

In these days of electric heating it occurs to me that the comfort of the astronomer might be secured by a special body pad, heated by electricity, which would keep the temperature immediately surrounding the body of the astronomer at 98 deg. fahr, so that there are no rises of temperature and no currents of air ascending and going out of the normal dome opening except those which the astronomer may cause, his body being at a temperature of 98 deg. He should, therefore, surround himself with a narrow electric heating pad which would keep his temperature, and not increase the bulk of heat around the dome. If this were done, I think a material reduction of the unequal temperature, and the difficulties which are consequent upon that, could be secured in the Hartness design.

THE AUTHOR. The discussion of this paper was opened by Mr. Warner with a reference to some historical events of a more or less personal nature. Perhaps a similar beginning for the closure may be fitting.

About thirty years ago the author had the pleasure of looking in through an open door into the works or laboratory in which the great Lick telescope was being built. The works was owned by two young men who designed this telescope and who could be seen in among the workers directing the construction. Their qualities as men and engineers must have contributed to the depth of impression and the pleasure gained by merely looking through the open door.

The laboratory was not mammoth in size, but was large enough for this great undertaking and for the greater accomplishment of building a grand reputation for these young men. That the credit was fully merited has been proved many times since by the continued progress and greater achievements, so that through these years they have stood preëminent as designers and builders of great astronomical observatories and instruments. These two young men of thirty years ago have honored the author by discussing this paper. Mr. Swasey's name follows Mr. Warner's, just as it has stood in the firm name of their organization. But there has never been a more satisfactory example of equality of partners than is exhibited in this ideal firm.

The discussion of Messrs. Warner and Swasey is followed by that of Dr. Kent, Mr. Hess and Dr. Hutton. The eminence of these gentlemen transformed their discussion into the most important part of the paper, and leaves little in the way of reply, except to clear up a few points that were left obscure by the description.

Mr. Warner calls attention to the greater weight of the moving parts of the turret telescope, and the probability that the main difficulty would be encountered in an attempt to provide a reliable driving clock.

Thus far, the need for greater precision than that furnished by the simple motor drive has not been reached. If, however, the greatest precision is desired, it can be readily supplied by the driving mechanism that is used by Prof. E. C. Pickering, director of the Harvard College observatory, for driving his large telescopes. This mechanism was designed by Willard P. Gerrish, and it is of such a nature that it has a great reserve of power. The main drive is an electric motor which is under the control of the clock mechanism. Its scheme of operating is most favorable for meeting any ordinary variations in load.

Like all other schemes, it includes an auxiliary slow motion by which the advance of the telescope may be accelerated or retarded to offset the effects of refraction and other elements that tend to change the position of the object in the field of the telescope. This device would certainly furnish a drive of great precision for any load, or variation in load, that would be found in moving the turret. The variation in load, of course, would be due to change of frictional resistance, and this would not be very great so far as the frictional resistance of the rolls on which the weight of the turret is supported, is concerned. It would be mostly in the bearings of the transmitting gears that carry the motion to the driving rolls.

The turret with its telescope and counterweights weighs about 2 tons, but it rests on an ample seat of 7 ft. in diameter, and it is not delicately poised. It seems to the author to be more reliable than the less stable mounting of the standard telescope with its lighter load.

Mr. Swasey calls attention to the sheltering effect of the standard telescope dome in its protection of the instrument, and thus delicately calls attention to the obvious exposure of the tube and the object glass of the turret telescope to the gales as well as milder winds. The present instrument is exposed to a clear sweep of wind from the southwest to the northwest. Since its erection we have had gales, but none that has wrought destruction to any structures in this locality.

The position of the tube when it is not in use brings it nearly horizontal, and may be about 2 or 3 ft. from the ground when pointing east. In this position it could be readily provided with a stall or separate shelter into which it could be securely locked, but there has not thus far seemed to be any real need for such protection.

If the object glass should be still further protected, it could be disconnected from the tube at this point and deposited in a box, which could be tightly closed to protect it from moist air, and yet keep it at the temperature of the outside air.

Regarding the protection of the tube from wind pressure while in use, the author has contemplated providing a shelter in the form of a sheet metal jacket or outer tube, which would be clamped firmly to the massive turret and stand clear of the telescope tube. If this outer tube should have a diameter of about 1 in. greater than the telescope and perhaps a length of 1 ft. or more, it would provide an ideal protection. It would be pivoted to the turret concentric with the declination axis, and would be free to follow the telescope when

it was being changed in position of declination. After that it would be firmly clamped to the turret, standing clear of the main tube. But all this may never be applied, for barring the most exceptional winds, the bracing of the tube and the exceptional mass and stability seem to hold it firmly and beyond any vibration that may be detected in ordinary work.

The writer waited over half an hour in an extremely squally wind one evening before he could see the star move from the cross wire, and then at an exceptional blast it moved off for a second or more and then back to its position. He had selected a star that would bring the telescope into the most unfavorable position for stability; namely, with the wind striking squarely at the side of the tube and its well sweep. In this position the tube depends solely on the control of the two worms on opposite sides of the 24-in. declination wheel.

This windy evening would undoubtedly have been a poor "seeing" time, even if the tube had had a perfect shelter. During the whole time there was probably a hazy elongation of the star in the direction of the wind, such as would be due to a slight quiver. The author did not see it then because he had adjusted the focus to illuminate the cross wires. He has seen it since in extreme wind when he had the focus adjusted to make the object sharp.

One more point regarding freedom from vibration should be stated. The tube is short and it is rigidly braced as to right ascension position. Its structure, taken as the combination of its large declination axis and its rigid well sweep brace is one that does not readily vibrate.

The subject of patents and their effect on the advancement of science which has been discussed by Messrs. Kent and Hess is too great a one to be fully treated in this paper. It is hoped, however, that this discussion will bring out a paper wholly devoted to it, for engineers should contribute to public opinion on a subject of such vital importance to the economic welfare of this country, as well as to the advancement of science.

Dr. Hutton's reference to the adverse effect of an unequal distribution of heat in the turret casting is an important point that should be borne in mind in times of greatest extremes of temperature inside and outside of the building. If its effect is appreciable, it could be readily tabulated for various positions of telescope and various temperature differences. In this work thus far the author has not found an error that could be directly traceable to that cause. The statement reflects very little light on the subject for it depends on the precision of his work.

Regarding the accuracy of the telescope in finding faint objects, he wishes to say that one authority of note, who spent an evening with the telescope, told him that he thought there were very few equatorials in the country more accurate in their performance, and although this gentleman's experience was based on only one evening's observation and could not take into account the effect of variation in temperature, the writer feels that it may indicate the reliability of the instrument, and that a great change on cold nights of 20 deg. fahr. would have been noticeable to him.

There is food for thought along the line of Dr. Hutton's suggestion for an alternative scheme of providing electrically heated clothing for the observer who is to be exposed to the low temperatures in the standard observatory. If this could be carried out without objectionable features, it certainly would be a great boom, but the writer thinks, as already stated, that the turret type of observatory has a stability, convenience and reliability of performance that should make it a preferred type, independent of its comfort providing qualities.

No. 1326

EXPENSE BURDEN: ITS INCIDENCE AND DISTRIBUTION

By Sterling H. Bunnell, New York

Member of the Society

As the various problems of engineering design and construction are reduced to the application of known rules, precedents and data, the element of cost enters more prominently into the calculations on which the choice of plans is based. Almost any engineering construction, whether railroad, bridge, tunnel, transmission line, factory or mechanical device, can now be produced with materials and methods in easy reach. The question, "Can it be done?" has given place in importance to "Will it afford an attractive return on the investment?" The engineer must now master in addition to the mathematics of materials and of electricity, the problems of financial operation, the principles of estimating correctly and providing for fixed charges as well as operating expense, and all the other details of accounting required in the continued successful operation of an enterprise after construction is finished.

2 By far the most difficult of the problems presented for joint solution by the engineer and the accountant is that of the distribution of factory burden on correct principles. The original conception of burden treated it as a wholly unfortunate, objectionable and regrettable outlay, to be kept as small as possible, and subtracted from the gross profit of the factory operation to obtain the figures of net profit. Every possible effort was made to charge not only wages and material, but also repairs and supervision, in some way or other into the particular product that happened to be under construction. The residue of expense which could not be disposed of arbitrarily by this method was collected into a single expense account and distributed by proportion over the cost of the products. The usual method adopted for carrying out the calculation was to distribute the expense

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

either as a percentage to cost of labor, or as an hourly charge to be added to the wages cost, of every hour's labor performed. Where this conception and practice exist, every bill that happens to be paid in a given month, or at the best, every charge incurred during that month, is regarded as expense to be distributed over the product of that month. If something breaks down and is repaired at heavy expense, the goods cost more that month because of making the repairs at that particular time. Accounting of this character has made it possible to operate many a factory at a huge profit, until the economically worn-out shell could be unloaded on some capitalist innocent enough to dispense with engineering advice, and purchase on the book figures of past operation.

3 The basis of this scheme of distributing burden by scattering it as evenly as possible over everything in sight, is the wholly erroneous conception that all the operating expenses of the factory can be divided into two classes, productive and non-productive, the first including all useful work on the manufactures to be sold later, and the second all waste, a dead loss to the organization. The logical development of the theory teaches that the ratio of non-productive to productive expense should be kept as low as possible; that the best manager is he whose expense ratio is the lowest; and that increase of expense ratio by high-powered machinery, trained helpers to save the time of skilled workers, and liberal outlay for good tools and their upkeep, cause loss. The absurdity of the conclusion and therefore of the premise, is evident. No legitimate expense is truly non-productive and some other definition must be found for the expense burden in order to indicate its true significance.

4 Every task performed in a shop under good management is directed toward accomplishing quickly and cheaply some definite and useful end. The purposes served by the various details of the work differ widely. Some of the tasks are performed directly on particular pieces of the factory product and are properly called direct. Others no less productive are on miscellaneous lots of pieces, like cleaning castings in the foundry; or on several dissimilar pieces at once, as in operating a group of automatic machines. Others have no direct contact with factory product, as in repairing belts, operating cranes and sweeping floors. All of those tasks which cannot be conveniently split up, measured into portions and charged direct to particular pieces of factory product are properly described as indirect.

5 The ratio of indirect to direct expense is no indication of the efficiency of the management, except as between two precisely sim-

ilar operations. In fact, a high ratio of direct to indirect may indicate extreme inefficiency, exactly the opposite of the accepted belief, and this under widely differing conditions; for instance, with a badly managed shop where every man charges his full time direct to a product, and runs his own errands or serves as his neighbor's helper on occasion; or, on the other hand, with a factory equipped with automatic machines under the care of skilled men whose time is charged direct to the cost of product.

6 Experience gained in connection with the cost accounts of factories and shops covering a wide range of manufactures shows ratios of indirect to direct varying from 20 to 150 per cent; and even higher percentages would not necessarily be surprising or discreditable to the management. The introduction of scientific management always increases the ratio of indirect to direct expense, and yet decreases gross cost as well as prime or direct cost. In fact, as between a shop under the best methods of twenty years ago and the best modern practice, prime costs afford no possible basis of useful comparison. High indirect and low direct costs are to be expected with modern equipment, even without scientific management. Heavy cuts by a powerful machine tool in charge of an efficient semi-skilled man at usual day wages often involve an operating cost of $1 per hour for the machine (an item of the expense account), against a direct wages cost of 30 cents for the man, a ratio of 330 per cent. Whenever, as in this case, the direct cost is a mere fraction of the expense burden, to keep accurate costs of the labor and material items only is nothing less than absurd.

7 Scientific management, while increasing the ratio of indirect to direct expense, decreases direct expense in greater proportion. The net saving cannot therefore be ascertained except by accurate knowledge of the details of each class of expense. This knowledge must be continuous, always at hand for instant use. The works manager cannot wait twelve months for an annual audit to show whether a new planning department has saved its cost, or increased operating expense has been followed by a greater increase in value of output. The engineer who is trying to persuade a superior officer in the financial department to authorize the purchase of newer equipment or the increase of the supervisory force will appreciate the advantages of accounting methods by which the exact results of changes in ways of doing work may be demonstrated through the factory accounts. In every factory, the exact result of each new task-setting operation, change in method, and improvement in equipment, not only in regard

to the wages and supplies of today, but also with respect to the operating expense accounts of the month and the fixed charges of the year, should be known in detail and at once. These requirements can be met only by a clear and comprehensive analysis of the true incidence of the fixed charges, and of the indirect details of operating expense, followed by the introduction of methods by which each item of factory product may be charged with that proportion of the indirect expense which corresponds to the cost of maintaining the shop facilities used in performing the manufacturing operations upon that item. The same degree of precision obtained in charging to the cost of work the direct expense for material and labor, should be reached in charging the large and important indirect expense.

8 The part of the engineer in developing the scheme of correct distribution of burden is much more than mere scientific interest in the work of some outside accountant.

9 The first step in the solution of the problem is to analyze the data and conditions. Of the items which go to make up indirect expense or factory burden, some of them are observed to have a very clear incidence upon definite points within the factory. Consider the annual interest on the value of a factory building. Surely it will not rise with an increase in wages or cost of material; nor with an increase in output; nor does it concern anything which takes place in some other building. The building is there to serve a useful purpose; and whatever benefits by that purpose should pay its share toward the interest on the cost of the building. The purpose is evidently the housing of machines and their operators, and each of the machines forms a unit in the productive scheme of the factory. Let the building be divided accordingly (by imaginary lines) into productive units, and require each to earn the interest on that part of the whole building which is occupied by the equipment, operator and work in progress, of the unit.

10 Once attacked from this point, the problem of a correct distribution of expense burden becomes easy of solution. The first step in the process is to obtain a schedule of annual fixed and operating expense. This includes the reserves for interest and depreciation, taxes, insurance and other fixed charges, a reasonable allowance for the expected average repairs, and the expense for supervision, small tool upkeep and power cost. Each of these charges is to be split up and apportioned among the particular productive units to which it belongs, the total to be carried as an element of the cost of that part of the factory product which passes through that particular produc-

tive unit during the year. The incidence of each detail of the burden being clearly on the productive unit, which comprises equipment, accessories and a portion of the building and land, the whole of the annual charge has as direct an incidence on the work passing through the production center in the year as the wages of the operator on the equipment of the center. The labor cost is the amount paid the operator, on day, hour, piece, premium, bonus or whatever system is used. The equipment cost is a suitable fraction of the annual burden. As most of the burden elements (interest, taxes, etc.) have a time factor, it is convenient to reduce the annual charge to an hourly rate, and to make the charges to cost of work on the same time units that are provided by the work records of the operatives.

TABLE 1 SCHEDULE OF EXPENSE[1]

Interest on Land and Building	$50.00	Interest on Equipment 5 per cent on half new value	$168.25
Depreciation 2½ % on $1000 Valuation	25.00	Depreciation per schedule	407.00
Taxes 1% on $1000	10.00	Taxes 1% on assessment	39.00
Insurance, per year	5.00	Insurance	35.00
Heat and Light. Share of this Building	50.00		
Building repairs estimated at	25.00		
Total Building Charge	$165.00	Total to Distribute to Equipment	$649.25

11 In the practical application of this fundamentally correct system of apportioning expense burden, several interesting and illuminating conceptions have been brought out. The labor of calculation is reduced by following a standard procedure, commencing with the schedule of expense, Table 1. The items belonging to the land are first grouped, and divided by the square feet of occupied land to obtain a land factor in dollars per year per square foot of land. Next, the portion of land belonging to each building is tabulated, and the area of each portion in square feet is multiplied by the land factor to give the total land charge for each building. This forms the first element of burden for the building. The interest, taxes and insurance are calculated on the book value of the building; the annual cost of heat and light are computed and the items totaled to obtain the annual building factor, and divided by the floor area

[1] A unit value of $1000 for land and building and a new cost of $6730 for equipment are taken as a basis for these estimates.

TABLE 2 DEVELOPMENT OF BURDEN CHARGES

Equipment Number	No. of Machines	Description of Equipment	Equipment Appraisal: New Value: Each	Equipment Appraisal: New Value: Total	Equipment Appraisal: Life: Age	Equipment Appraisal: Life: Future	Building Factor: Sq. Ft.	Building Factor: Charge	Power: Estimated H.P. Indiv. Motor	Power: Estimated Running Time	Power: H.P. Ratio	Power: H.P. Hours	Power: Charge	Share of Accessories Value: Motor (on H.P.)	Share of Accessories Value: Small Tools and Grinders: Ratio	Share of Accessories Value: Small Tools and Grinders: Charge	Share of Accessories Value: Shafting and Furniture (on Sq. Ft.)	Share of Accessories Value: Present Value of Equipment Total Value Accessories	Interest Taxes and Insurance (on Total Equipment and Accessories)	Depreciation	Repairs: Ratio	Repairs: Charge	Total (Columns 9-14-21-22 and 24	Foreman and Helpers	Total Annual Burden Cols. 25 and 26
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27
1634	1	Engine lathe, 16 in. by 6 ft., 8 in. chuck	$350	$350	10	10	45	$ 6	2½	2400	6000	2480	$29	$27	7	$25	$15	$175	$67	$14	23		19	91	135	226
1635	1	Engine lathe, 13 in. by 5 ft., 6 in. chuck	300	300	10	10	60	8	2	2400	4800	1990	24	18	7	25	20	150	63	12	20		16	80	135	215
1636	1	Milling machine, 6 in. vise	600	600	10	10	138	17	2	2000	4000	1650	20	18	10	36	46	300	100	24	39		33	133	112	245
1637	1	Driller, 10 in., 15 in. swing	125	125	20	10	61	8	½	2400	4200	493	6	4	5	18	20	42	42	5	8		7	34	135	169
1638	1	Driller, 11 in., 24 in. swing	175	175	20	10	77	10	2	2400	4800	1980	23	18	5	18	25	58	61	7	11		10	61	135	196
1639	1	4 Spindle drill	200	200	20	10	63	8	1	1000	1000	413	5	10	5	18	21	67	49	7	11		42	42	56	98
1640	1	1½ Milling machine	530	530	10	10	99	12	2	1800	3600	1480	18	18	10	36	33	265	87	21	34		29	114	102	216
1641	1	Screw press	50	50	10	10	16	2	..	..	..	..	..	..	..	..	5	25	5	1	3		3	9	..	9
1642	1	Shaper, 15 in. stroke, 8 in. vise	330	330	10	10	16	2	2	1800	3600	1490	18	18	3	11	5	165	34	12	19		18	69	102	171
1643	1	Milling machine, small	350	350	40	5	30	4	¼	1000	250	103	1	2	6	22	10	29	34	4	12		19	40	56	96
1644	1	No. 1 Universal Grinder	250	250	15	10	36	5	2	100	200	83	1	18	4	15	12	100	45	8	14		14	42	6	48
1645	1	Screw machine	550	550	20	10	25	3	2½	500	1250	501	6	26	7	25	8	183	59	14	23		30	76	28	104
1646	1	Cutter grinder	150	150	20	10	30	4	¼	100	25	10	....	2	3	11	10	50	23	4	7		8	23	6	29
1647	1	Surface grinder	150	150	10	10	30	4	⅝	1000	375	155	2	4	1	4	10	75	18	6	9		8	29	56	85
1648	1	Surface grinder, small	125	125	10	10	20	3	1/16	500	31	13	....	2	1	4	7	63	13	5	8		7	23	28	51
1649	1	Polishing lathe	30	30	20	5	35	4	⅛	500	62	26	....	2	3	11	12	6	25	2	6		2	14	28	42
1651	1	Small forge and equipment	40	40	10	10	35	4	¼	750	188	78	1	2	1	4	12	20	18	2	4		2	13	42	55
1653	1	Power hack saw	40	40	10	10	35	4	¼	500	125	52	....	2	2	7	12	20	21	2	4		2	12	28	40
1654	1	Speed lathe, 12 in. by 6 ft	75	75	10	10	48	6	⅛	300	38	16	....	2	4	15	16	38	33	4	7		4	21	17	38
1655	1	Lathe with collets and turret attachment	300	300	10	10	66	8	2	1000	2000	825	10	18	10	36	22	150	76	13	22	Repairs apportioned by judgment, and include small tool repairs	16	69	56	125

1656–1659	4	Rivet bench lathes	450	1800	10	10	92	12	⅛	4800	600	250	6	9	32	116	31	900	156	64	105	98	285	317	602
		Bench space for two men		100	10	10	255	31	..	4800					12	43	83	50	126	11	18	5	65	270	335
		Totals		6620			1312	$165			34,144	14,088	170	220	138	500	435	2931	1155	242	407	361	1345	1850	3195
		Accessories																							
1650	1	Wet grinder, 12 in	80	80	10	10	34																		
1652	1	Grindstone	30	30	10	10	61																		
		Shafting $180; miscellaneous trucks, etc $200																							
		Storage space					62																		
		(Total floor area of shop)		6730			1469																		

[1] Value of land and building occupied by shop equipment in column 3, $1000

Interest 5%, Depreciation 2½%, Tax 1%, Insurance ½%	$90
Building repairs, estimated at	25
Heat $40, Light $10	50
Total building charge	$165

of the building to give the square foot factor. The next step is a map or diagram showing each production center with its working floor space, including in the list of centers erecting floor spaces, benches, and all equipment on which men are employed or operations performed on factory product. The centers are then to be tabulated in order by departments as in Table 2, with the appraised value and square feet of floor space occupied by each. Multiplying the latter by the square foot factor, the total building factor of each productive unit is obtained.

12 Accessory equipment goes with the producing machine of each production center, that is, shafting, belts, all or a share of a motor, small tool equipment, and the like. The value of such equipment is generally much less than the value of the principal machine, so that absolute accuracy in apportioning accessories is not essential. If the value is very small, it may be divided among the units in proportion to the value of their producing machines; if large, shafting and belts may be apportioned to the units on the basis of the floor space occupied by each, motors driving groups on the basis of the working horsepower-hours of each productive unit, and small tools by judgment. If the greatest accuracy is desired, the accessories may be treated separately and each one measured, appraised and assigned to the productive units accordingly.

13 The remaining calculations are carried out for each unit independently, including the value of the accessories with that of the main machine. A depreciation rate is set for each unit, or class of machines, and the corresponding annual charge is computed. Interest on the value of the machine, taxes and insurance are also provided for. All these are definite quantities, not open to argument. There remain certain items not as definite: power cost, repairs, general labor, factory supplies and (in a machine shop) cost of tools, each of which requires special treatment. Power, if it could be metered separately to each productive unit, would become a direct charge; but as this is impracticable, and the power cost is not of the greatest magnitude, a reasonable approximation can be made by estimating the average power consumption, the average running time, and so the average annual horsepower-hours of each unit, and multiplying by the cost of a horsepower-hour to obtain the power factor of the burden charge. Supplies and general labor are best divided by estimate and judgment. Repairs to the individual units cannot possibly be foretold; but the average total repair cost of the plant, or even of the several departments of the plant, can be estimated with some degree of precision,

and distributed to the productive units in any of several ways. Perhaps the most practicable is to set by estimate figures representing the probable proportion of repairs likely to be required by each kind of machines, placing 1 opposite the machine likely to require the least expense for upkeep, 2 for those machines likely to require twice as much expense, and so on. By adding these figures for a denominator and using each figure separately for a numerator, the total estimate for repairs can be divided up among the several units. There is no advantage for cost-keeping purposes in attempting to charge the actual cost of repairs into the operating expense of each unit. In repairs for lightning and the like, which generally become necessary without warning, there is no reason why the expense burden and therefore the cost of work done by one of several similar machines should be temporarily increased merely because that machine suffered breakage and was repaired at this particular time. Standard power and repair costs are a great help in rational cost-keeping, since they minimize fluctuations in cost not due to difference in management. By opening separate repair accounts, charging these accounts with actual cost, with all its momentary variations, and crediting the standard estimates, the varying balances tell an interesting story of the work of the repair gang, and keep the fluctuating costs of repairs from exerting a disturbing influence on the cost of product. A power-plant account similarly operated is also very useful.

14 The tabulation now includes subdivisions of all the expenses which go to make up the factory burden, up to the point at which the completed goods leave the factory, but exclusive of the administrative expense. If the total burden chargeable to each unit is divided by the respective annual working hours of the units running full time, an hourly rate for each productive unit will be obtained, which if applied to the cost of the work done by each unit, will in a normal year of full working time accumulate a credit which will balance the shop expense burden. There are, however, many lost hours in the course of the working year and such losses reduce the credit which is to stand against the annual shop burden, and tend to produce a deficit. Short time may be due to any of three causes, each of which gives to the resulting deficit a different significance. In the ordinary operation of a factory, time is lost by the productive units through illness of operatives, changes in the force, breakages and repairs, and lack of capacity in other units. Obviously, a deficit in the credit against burden due to losses of this character is a manufacturing expense, and should be distributed to the cost of work.

15 Lost working time, however, may be due to sales department conditions, as when machinery is installed for the manufacture of goods which have a seasonable demand, so that they can be sold during part of the calendar year only. A deficit due to lost time of this character is selling expense, and not part of manufacturing cost. In such a case, the sales department undertakes to earn through profit on the sales the necessary amount to pay for the use of the capital invested in the equipment over the period of idleness. The manufacturing cost cannot be increased merely because consumption ceases after a time, so that the machines have to be stopped. If it costs $1000 to operate equipment for three months, and another $1000 for the following three months, it is not to be supposed that it costs $2000 to operate for the first three months only. Factory product can be made continuously, stocked up during an idle period, and sold at the proper time; if such a course is inadvisable, by reason of tying up capital, or loss of interest on the money represented by stored goods, or risks taken, the expense of intermittent operation falls outside of the manufacturing account. Admittedly, someone must pay the fixed charges for each period of idleness; but the party liable in cases of the kind supposed, is not the works manager, who would be glad to operate continuously, but the selling division of the organization, which must take the consequences of sales conditions. Correct accounting practice will not justify the distribution of selling expense to goods before they are sold; wherefore, manufactured goods should not be placed in stock with a burden allowance which includes selling expense, though it is quite proper to include all other items of burden.

16 The third possibility of the deficit due to lost time is that it is a consequence of bad trade conditions. Lost time of this character is neither manufacturing nor selling expense, but a business loss which should be made up out of the profits of good years. Conservative management accumulates a surplus to give stability to the rate of distribution to the stockholders, by reserving a portion of the earnings of good years to be used in maintaining dividends in times of depression. With correct methods in distributing factory burden, the loss due to short time operation, usually buried out of sight among other expenses which have no relation to running time, is clearly shown, so that an appropriation may be made from the surplus account to carry the deficit of poor years by the extra earnings of good years.

17 The divisor used on the annual burden of each unit should therefore be less than the full annual shop hours, at least by a suit-

able allowance for holidays and other lost time. In the case of a large special machine for which there is work only part of the time, only the expected working hours should be used as a divisor, for such a unit to be profitable must earn its annual charges in the running time which can be given it. Whenever conditions change so that such a unit can be operated a larger proportion of the time, the hourly charge should be correspondingly reduced.

18 There remain some items like salaries and office expense, the incidence of which cannot be traced to definite productive units. This class of expenses, however, is incident to the operation of the factory as a whole. The larger productive units, with their greater capital value and heavier operating expense, involve a greater tax on the management and administration than the smaller units. The burden charges belonging to the units provide a very fair measure of the responsibility of the management in connection with each, and serve well as a basis of distribution of the overhead charges. It is proper, therefore, to express the overhead expense total as a percentage of the total burden distributed to all the productive centers together, and to raise the hourly rate of each unit by the same percentage. Thus, if $100,000 is distributed to centers, and there is an undistributed overhead of $10,000, the rates can be raised 10 per cent, which will provide for carrying the whole $110,000.

19 Each productive unit of the factory is thus valued at an hourly rate which is to be charged in the cost of each operation performed by the unit. The total cost of an operation accordingly consists of a labor charge and an equipment charge. The basis of each machine rate is a careful investigation of the operating costs of the particular machine, so that the rates set are in accordance with the facts in each case. Whatever fluctuations there may be in operating expense are likely to balance each other above and below the calculated charges. But as there will be some variation in any event, it is permissible to obtain the advantage of easy calculation by equalizing the machine rates into two, three or more round figures, each covering productive units whose exact rates fall near together. On the job-tickets by which the time or piece price of the labor is computed, a space should be provided for the burden figure, so that the completed job-tickets will give both elements of the cost. Any change in methods or men will now indicate its complete result by a comparison of the job-tickets for the same work done under former and present conditions.

20 The accounting detail of the expense burden distribution is simple and effective. A burden distribution account is opened, to

which is charged the calculated amounts of Table 1. The account is credited with the total of burden charges to cost of work, by weekly or monthly periods. The balance indicates the loss by failure to utilize the full capacity of the factory; while a credit balance, if one should occur, indicates a condition of unexpectedly good management, or good luck, by which the factory has worked to better advantage than was believed probable. A standardizing account of this kind is very useful to the management. All the variable expense accounts can, if desired, be handled as suggested for power and repairs, so that uniform credits offset the varying debits, and the balances afford gages for the efficiency of the control of the departments originating the accounts.

21 A constantly increasing balance in one of the expense accounts will occur if there has been a permanent change in policy with respect to this item. In such a case, it will be necessary to change the equipment rates affected by this change. While it is not at all difficult to go over a column of the tabulation and revise the figures, it will generally be sufficient to change the rates by proportion, enough to absorb the difference. But it is not necessary to distribute all the burden by machine rates. The overhead of salaries and office expense, the cost of storage and handling, and, if desired, any small difference in burden actual and distributed, may be distributed as a rate per pound on a homogeneous product, or as a rate per order if orders are all of the same size, or as a percentage of gross cost, if that seems the most reasonable method. In every factory, careful analysis of the facts will point out the true incidence of expense due to factory operation, or factory product as a whole, rather than individual production centers.

22 A valuable result of localizing variations in expense burden is stability in the cost of the manufactured product. Because a repair force is discharged to save expense, and everything is allowed to fall into bad order, is no reason why the cost of goods made during the period should appear to decrease. The management is in effect borrowing part of the cost from the stored energy of the organization, or discounting the future. All fluctuations not directly due to a change in manufacturing methods are objectionable if carried into cost of product. Where the cost of similar articles varies uncertainly from day to day, each fluctuation is likely to have a different cause, much time is taken in searching for the causes, and incorrect conclusions are often drawn, or it is assumed that some unknown and unusual condition is responsible, and the attempt to trace the cause is given

up. Increased labor cost is easily discernible and can be traced directly to its origin. Variations of material cost are quite as easy to explain; but where burden charges vary with labor charges, under the old percentage or hourly rate methods, a change in cost of labor involves a totally unrelated change in the burden charge, and no clear conclusion can be drawn.

23 Exact calculations are at the base of all well-planned engineering work. Exact uniform methods of cost calculation should be at the base of all factory accounting systems. Wherever associations of manufacturers operating on similar work have investigated the cost-keeping methods of the various factories, the investigators have been surprised to note the wide variation between the apparent or book cost of the same products, made under the same conditions, but figured in different ways. The inevitable result is that some factories bid for work below cost, and thereby injure the business chances of all. The remedy is not the maintenance of unduly high prices by agreement, thereby inviting competition from every outside shop; but an exact analysis and apportionment of the expense burden in such a way that each item of factory work will be properly charged for the use of that portion of the shop equipment and facilities which is devoted to it, and the true margin of profit between cost and selling price will be clearly apparent.

DISCUSSION

H. F. Stimpson.[1] It is natural to discuss a paper from one's own standpoint, and mine is that of the engineer rather than that of the accountant. Hence I shall not attempt to pass upon the technical details of the process. I feel justified, however, in discussing the relation of accounting to business in general.

In the performance of an act there are four principal steps:

a The issuance of the impelling order which involves
- (1) a conception of the desired result
- (2) a perception of the available resources
- (3) the formulation of a working plan
- (4) the compilation of competent directions
- (5) the transmission of the directions

b The performance of the act.

c The recordance of the act, which is the function of the accountant.

[1] Consulting Engineer, New York.

d The determination of the ratio between the opportunity expressed in the directions and the result as shown in the accounts. This, as Mr. H. F. J. Porter has so aptly put it, is the efficiency of the performance. Its determination is an executive function.

Any accounting whatever is, therefore, only the third link in the chain. The account may be accurate to the last degree; the act which it records may be bad, inefficient, wasteful and extravagant. The accuracy of the record is no evidence of the propriety of the act.

Accounts are of value only when they are efficiently used. To this end, competent and specific directions or descriptions of opportunities are essential. If these have been issued to the worker and are backed up by such personal explanation as is necessary to his understanding the performance of the act itself, the supreme end in view, will be much more likely to be highly efficient. An ounce of prevention, in the shape of competent, specific directions, is better than a pound of cure. Our executive and administrative methods are short in foresight and long in hindsight. This has been shown over and over again in the tremendous variation between possibility and performance which efficiency engineers have repeatedly brought to light by the application of knowledge which was just as available before the act as afterwards. The accurate and painstaking work of the accountants had, however, never revealed it.

In one of my recent investigations, I demonstrated that only 13 per cent of the operations were competently and specifically directed and only 33 per cent were competently recorded. For lack of proper directions the cost of the excess of 20 per cent of accounts over directions was largely a waste of time and money, yet neither was at all adequate.

In a recent address before the School of Commerce, Accounts and Finance of New York University, I said that when an accountant approached the problem of devising a system of records or of auditing an existing one, he should first call for the directions which had been given and confine himself to paralleling these with records. If such records did not give adequate information he should request that directions, initiative of the acts of which information was desired, be formulated for him as a basis for his own work. This is an engineering and not an accounting function, and requires a training and power of initiative which the accountant does not possess. Facts should be the basis of our directions and the same facts, subdivided in the same detail, should be shown in our accounts. This

process should begin with the directions which the general manager receives from the secretary, in the form of the working resolutions passed by the directors. The records of the treasurer should exactly correspond with such directions, as to their character and scope. When they do so, a determination of the efficiency of the management is possible. The general manager should formulate the directions for his working organization only after he has fully digested the advice of a complete and competent staff. The records of his department of accounts should accurately correspond to such directions in every detail and be modelled thereupon, instead of the reverse, as is usually now the case.

The efficiency of the acts should be principally dependent upon competent prospective direction, rather than upon a retrospective contemplation of accomplished performance. It is better to pay the safe maker than the detective.

The chief function of accounting should be the detection of flaws in the execution of our plans rather than the original formulation of the plans themselves. The proper basis for that is the research work of a complete competent technical advisory staff. The proper function of the general manager is the digestion of such advice, the formulation of complete competent directions and their transmission to the worker.

BENJ. A. FRANKLIN.[1] The main principle of the method of placing expense burden, suggested by Mr. Bunnell, was in operation in the plant where I first became acquainted with the subject of costs, some twenty years ago, viz., to discover rates per hour for each machine, which, when charged to production in the proportion of hours of machine operation, would absorb the indirect plant expense.

As a theory it is very difficult to refute in very many cases. It is of course evident that, disregarding the labor element, it is more expensive per hour to operate, for instance, a $5000 machine tool than a $1000 one, certainly as far as the power it uses and the depreciation charges are concerned, but dependent upon the work, the $1000 tool may use more crane power or more cutting tools.

Of course this expense burden is the bête noir of the cost system. Changing conditions alter its aspect constantly, and a practical point which it seems to me should not be overlooked is which method of placing expense on the saleable article is the simplest, and the easiest

[1]Director, Strathmore Paper Co., Mittineague, Mass.

to carry to constant successful operation, and which will meet the exigencies of competition and present to the executive a proper knowledge of the moving situation for shop control.

It is difficult to get manufacturers to use methods that demand any adjustment, or much thought and labor in maintaining. An examination three or four years ago of some eighty plants using machine tools showed that a very small percentage used any method at all of applying the expense burden in their costs. In competitive business it would seem more important to adopt a method of expense burden that is uniform and simple and understandable enough to be accepted and adopted by all, than to urge a method more theoretically right but difficult to maintain. The Cost Committee of the National Machine Tool Builders, appointed to devise a standard cost system, with which I worked, deliberately suggested, while it had under consideration at that time the machine hour expense burden, the choice of the man-hour or the productive hour wage cost as the basis for dividing expense, because it felt that one of these two methods alone would stand any chance of being accepted in any appreciable number of cases, and that if all adopted either the competitive result would be satisfactory.

To consider more closely the value of the plan outlined by Mr. Bunnell, it seems that it has certainly one first value, viz., that any machine tool buyer in considering the purchase, or as the author puts it, any engineer suggesting the purchase, of machine tools might well utilize this plan to work up figures showing the comparative costs of the same work done on different machines as a basis for consideration of their values.

It is somewhat debatable whether a manufacturer is entitled to add interest on his investment to his costs unless he is intending to be content with a very small profit over his cost. And if he adds interest on his permanent investment why not on his floating investment?

I cannot quite agree that the miscellaneous undistributed expenses such as salaries, office expenses, etc., which in Mr. Bunnell's example amount to a possible $10,000, as compared with $100,000 centralized expense, should be distributed per machine hour cost. It seems this simply taxes the expensive machines more heavily. Nor do I think this part of the expense is properly met by the later suggestions that it may be divided by the pound of production or as a rate per order or over the gross cost. Why be so particular in application of the $100,000 and so careless with the $10,000?

It would be a difficult problem also to convince either the ordinary

manager, or the sales force that they should have charged against the selling expense any share of the manufacturing expense because the season or the times did not match their efforts.

The plan of a standard cost with a balance either for or against it at all times would seem to have some advantages if the balances were always watched, but executives get careless about these things, and the standard cost gets fixed in their minds as the real cost, with a danger of misleading.

In order to overcome properly the variations of expense burden due to such natural and fixed causes as variations in first cost and power use, the most practical thing has always seemed to be to divide the plant into departments and divisions, including as far as possible like conditions. This being done, it is a very simple proposition to make the divisions of depreciation, power charge, etc., on a fair basis to each department and division, show the actual productive hour expense burden for the month and for the period in each department and division, to which can be added the general expense productive hour burden for the month and the period. Such an act of accounting any bookkeeper or cost clerk can quickly learn. It presents the actual situation and it has always seemed to meet all practical needs of competition and economy consideration.

I know of one case where a near-sighted executive with the machine hour burden kept a large machine idle and a small one always overburdened with work waiting, because, if the work were done on the large machine it would cost too much on his books.

There are very many good points in the paper of Mr. Bunnell but the last word on expense burden has not yet been spoken, and it would seem to the practical man that no system of costs will be widely accepted that can not be operated with a minimum amount of clerical labor, difficulty of maintenance, and correction, and the fluctuation of which does not picture the plant's fluctuations, and allow of a quick interpretation by the executive.

H. K. Hathaway. It is very gratifying to note that it is being recognized that low ratio of indirect expense to direct expense does not necessarily indicate a high degree of efficient management; nor does the reverse indicate the contrary to be true.

The method of distributing indirect shop expense described by Mr. Bunnell is essentially that developed under the Taylor system of management, and is probably founded upon that method, especially in the manner of arriving at the machine rates.

Under the Taylor system we have, generally, two broad classes of indirect expense: shop expense, that is, the expenses a concern would have if it manufactured its goods but had no commercial department; as would be the case if the products were turned over to a separate selling organization. The other, which we call general expense, is that which we would have if we conducted a purely merchandising business, but did no manufacturing. These two statements that I have just made are only generally correct, as, of course, a concern doing only a merchandising business may have some of the expenses that are ordinarily included in the cost of manufacturing, and vice-versa.

The first, shop expense, is distributed on a basis of machine hours times machine rates for time employed directly on product.

The second may be distributed in any one of a variety of ways, no one of them being logically correct, as Mr. Bunnell points out.

The one particular point in Mr. Bunnell's paper that strikes me as being novel is his proposition for charging selling expense, as well as the excessive manufacturing expense due to dull times, onto the product at the time of sales, instead of when manufactured. This has the apparently desirable feature of showing up and immediately disposing of losses that may occur, whereas, if charged onto the product at the time of manufacture and the product remains in stock for some time before sold, the loss or decreased profit does not show up until long after it has been actually incurred. In the same way it would seem to avoid showing a profit on products manufactured during prosperous times in spite of the plant being operated at a loss when sold.

The only objectionable feature that I can see at this time is that assuming that all indirect expense is charged off monthly, as it should be, there might be in many plants periods during which shipments or sales might be comparatively small by reason of a considerable quantity of work being under way in the shop, but none of which it was possible to complete or ship during the month or months in which the bulk of the work was done. This objection may not be important, however, and may be outweighed by the advantages. I believe, however, that it would result in such sudden changes in the profits and losses shown from month to month, as to be rather unsatisfactory from many points of view.

What Mr. Bunnell has given us is so interesting, that it is to be regretted that he did not go more fully into the methods of distributing the indirect expense. There is, of course, a danger of cost system,

particularly in respect to distributing indirect expenses, becoming unwarrantably complicated and requiring more labor in carrying it out, than is justified by the use that will be made of the results.

WILLIAM KENT. Mr. Bunnell will no doubt admit that some of his directions, while generally applicable, are subject to exceptions under some circumstances. In connection with Par. 18, he says: "It is proper to express the overhead expense total as a percentage of the total burden distributed to all the productive centers together, and to raise the hourly rate of each unit by the same percentage. Thus if $100,000 is distributed to centers, and there is an undistributed overhead of $10,000 the rates can be raised 10 per cent, which will provide for carrying the whole $110,000." Suppose that the largest productive center of a concern making a miscellaneous product is a well organized screw factory, whose distributed burden is $50,000 out of the $100,000 for the whole concern. Its proper proportion of the $10,000 undistributed burden is $5000. Now suppose that in the following year no change is made in the screw factory, but that in the other department expensive changes are made, such as the manufacturing of a new product, the introduction of a planning system, etc., increasing the total distributed burden to $150,000 and the undistributed burden to $20,000. It would not be proper to apportion any part of the $10,000 increase of undistributed burden to the screw factory, increasing the apparent cost of the screws made, but it must all be apportioned to those departments for whose benefit this extra $10,000 undistributed burden was incurred.

HARRINGTON EMERSON. I welcome this able paper of Mr. Bunnell's. The whole subject of burden and burden distribution is so unstandardized that any public presentation of personal experience and solution has great value.

During the twenty years I have wrestled with the question of manufacturing costs, various axioms have emerged.

In the first place, system should always be subordinated to efficiency, which should be made the main element in all cost records as it is of far more practical importance.

Accidental costs have no great value. They belong to faulty system and should be replaced by standard costs and efficiency factors. These two give more accurate costs than any system of accidental costs.

The accountant's field lies in the verification of operating state-

ments, not in their invention. The experience of the accountant, invaluable in its proper sphere, does not fit him to establish either standards of operation, or secure efficiencies of operation, both of which are fundamental.

Factory costs begin when materials, labor and equipment charges are requisitioned for factory use. These requisitions are the line of juncture between accountant and efficiency cost-keeping. The accountant is interested in the requisitions, which are the basis for his many records. The efficiency cost-keeper uses the same requisition as the basic and fundamental unit for comparison with his standards and to determine efficiencies, which are the relation between actual and standard, both shown on the requisitions, one requisition for each item.

The actual costs for plant or for department are the predetermined standards divided by the plant or departmental efficiency factor. Standards appear in the requisitions, actual results are later entered on the same cards.

All operating costs fall into three main classes, materials, personal services, and capital charges. Economically there is neither logic nor gain in treating these differently. These classes are again divided into seven sub-classes: direct materials, direct labor, direct machine charges, power, maintenance, supervision, rent. The efficiencies of these sub-classes are easily determined and establish the efficiency of the plant or of the department.

The four indirect items, power, maintenance, supervision and rent, are reassorted into three burden charges, on material, labor and machines. The universal cost formula therefore becomes

$$\text{Cost} = Q(P+B_{\text{p}})+T(W+B_{\text{w}})+t(R+B_{\text{r}})$$

This applies to any item however small, the cost of making a pin, for instance, to the operating costs of all the railroads of the United States for a decade. In this formula, Q = quantity of direct material; P = price of materials per unit; B_{p} = burden on price of materials; T = time in hours; W = wages per hour of direct worker; B_{w} = direct wage burden per hour; t = time in hours of direct machine; R = rate per hour of direct machines; B_{r} = machine burden per hour. These costs may be either standard or actual.

As to each of the seven main classes of costs, three direct and four indirect, there are four efficiencies, price, supply, distribution, use. If P or W or R is above market rate, there is inefficiency of price. If Q or T or t is overabundant, there is inefficiency of supply. If high-

priced P or W or R is used when low-priced would have answered, there is inefficiency of distribution. If Q or T or t is wastefully applied, there is inefficiency of use.

These four efficiencies are in dependent sequence. In a given case, railroad time tables, let us assume an excessive cost of printing, efficiency of price, 90 per cent; an over-supply printed, efficiency of supply 80 per cent; thick and comprehensive time tables distributed where abbreviated local tables would have answered the purpose, efficiency of distribution 60 per cent; twice as many time tables taken by travelers as they really need, efficiency of use 50 per cent. The end efficiency is $90\times\ 80\times 60\ \times\ 50\ =\ 21.6$ per cent.

As to labor and machine cost, I have known efficiency of use to be as low as 1.25 per cent, owing to four subsidiary efficiencies in dependent sequence. The stroke of the machine was three times too long, its speed was only 30 per cent of what was proper, the feed was 1/64 when it ought to have been 1/16, and four cuts were taken where two ought to have been enough, resulting in, $0.33\ \times\ 0.30\ \times\ 0.25\ \times\ 0.50\ =$ 1.25 per cent.

This formula applied throughout permits bringing home to the person responsible every item of inefficiency wherever occurring; and when standards are established and reliable, immediate, adequate records show the degree of approach to standards, there is no excuse for any avoidable inefficiency. If the fireman is instructed to keep the steam gage at 160 lb. and the gage confronts him every minute of the time, he will generally maintain the standard. Similar reliable, immediate and adequate efficiency gages are provided for every worker up to the president or board of directors.

The Author. Special emphasis should be given to Mr. Stimpson's statement that accounting records should parallel the directions given by the management to the various departments. If these instructions are not clear, they should be made clear. If the management is directing along incorrect lines, it should change its directions. All of the orders and instructions under which work is carried on in the factory, should be controlled by accounting divisions. The factory order with its cost sheet serves as an illustration. Instructions are given to the factory to complete certain work, and the cost sheet shows the progress of the work to its completion.

The expense distribution system, described in the paper, embodies the various records of the operations of the factory as a whole. The several burden accounts, such as power, heat, light and repairs, pro-

vide for comparison of the actual cost of the items of expense as incurred, with the schedule or budget prepared as a standard. The success or failure of the man in charge of each department, like the chief engineer with respect to the power plant, is shown by the balance to the credit or debit of his department.

Mr. Franklin, in suggesting the adoption of a method of expense burden distribution simple enough to be accepted, rather than a method which is correct, is in line with the arguments which meet the accountant continually in his endeavor to introduce better systems of accounting in connection with factory work. Accurate bookkeeping always requires care; the careless bookkeeper has no trouble in putting figures in his books, because he has not set before himself the standard of perfection. It is absurd to use any method of distributing expense burden that does not put the burden where it belongs. If the object of the system is merely to close the expense account, a simple balance entry transferring the total expense to the profit and loss account is enough to satisfy accounting requirements. But the thing which is needed is a detailed profit and loss account, showing the true gross cost of each piece of work done, and the true net selling price received for the item of work. The same holds if not the individual costs and selling prices. Only the totals of the items making up a class of sales are required separately. In a factory making a diversified product, it is highly important for the management to know whether an extra large repair force and power cost in any one department are covered by the receipts of that department, or whether the factory is running in one place at a loss and in another place at a profit.

Any disturbance in existing conditions, such as a cut in prices in one line of product manufactured, may cause the net profit of the works to disappear; and the cost system should be capable of showing cause and effect in such a case, without requiring the management to go out with a shovel and clear away the debris to find the cause of the explosion after it has occurred and the previously profitable operation has turned out a heavy loser.

The addition of interest on investment in machinery, to the cost of work done by the machinery, does not compel the manufacturer to be satisfied with a small profit. The term profit is used to cover a variety of meanings. Mr. Franklin appears to have in mind here the profit measured by the difference between the gross manufacturing cost and selling price. If the factory is to pay dividends on invested capital, they should be earned by the equipment represented

in the investment. If $5000 worth of capital is represented in a boring mill, it should pay the interest on the $5000. Interest, which in this case is equivalent to dividends on the stock value represented, is entirely aside from the extra profit from which it is hoped either to compensate for previous losses, to increase the surplus account, or to pay extra dividends.

Interest on floating investment, however, should not be included in burden charge for the use of shop equipment. If the floating investment can be identified, as for instance the value of a stock of copper tubes kept for use of the condenser department, it would be fair to include a charge for interest on the normal tube stock, in the expense charge of the department. If, however, the stock carried is for the benefit of the works as a whole, and cannot be divided into portions belonging to the departments separately, then interest on the floating investment becomes a charge to the general profit and loss account, and is as indistributable as management expense.

It is not intended that all general expenses shall be distributed per machine hour cost. The $10,000 assumed in the table represents the expense of the shop or works management, as separated from the general administrative expense. The point made here is that the salary of the superintendent and his office force is fairly chargeable to the work which the superintendent and his force perform. This work is the maintenance of the factory, the repair and upkeep of machinery, the supervision of the workmen, and other details of factory management, and the value of the work done by the superintendent and the office force is represented by the total expense burden. If it costs $10,000 to supervise the expenditure of $100,000 for the maintenance of the factory operation, the smaller sum may be correctly distributed as 10 per cent added to the larger.

Executives are indeed careless about watching the figures which show the success or failure of their work; but this is not an argument for less figures, but for more watchfulness. Some managers watch nothing but their cash balance, and are unable to determine why the balance increases or diminishes. Others depend on annual statements to show them the progress of their affairs, though these statements are belated records of what has been done and forgotten months before. By all means let us give the manager a statement of results in all departments under his control, and insist that he shall watch these statements and act upon the information provided.

Mr. Hathaway is in full accord with the best accounting practice in approving the charging of selling expense into the product at the

time of sale. There is no justification for charging the cost of product with selling expense which has not been incurred. Product placed in stock is not yet sold, and the cost of selling it is yet to be paid from the profits realized from its sale. Product in stock should carry manufacturing cost only, and this should not be inflated by adding the loss due to slack production, when under unfortunate conditions the shop must be run at short capacity.

It is not intended that all indirect expense shall be charged off monthly, but only that the monthly schedule shall be written into the burden distribution account. In making up a correct statement of conditions, the indirect expense chargeable to work in progress should be taken into the account. If at a certain time there is an unusually large amount of unfinished work, or work finished but awaiting shipment, present in the shop, this work as far as the factory is concerned has received the benefit of the factory operation, and should carry its proportion of expense burden. It would be proper if work is sold, but held for delivery, to charge in the selling expense burden on such work in making a formal annual statement, but ordinarily the quantity of work which is physically present in the factory and yet has already been sold and so is chargeable with selling expense is very small.

Mr. Kent is correct in calling attention to the fact that overhead burden should be distributed to the departments concerned as far as possible before distributing it through the machine rates. Wherever it is possible to differentiate the overhead expense belonging to departments, each department should be charged with its own share of expense. In connection with distribution of superintendence charges as outlined above, the salary of the foreman of Department *A* is charged into the expense of that department. If Department *B* requires an extra cost-keeping force, due to the character of its work, the cost of this force should be charged into the expense of that Department, and so forth.

Mr. Emerson, with his well-known power of analysis, has reduced the principles of cost-keeping to a formula which is clear and concise. It should be noted that some of the quantities in his cost formula can often be added together without changing the total; for instance, the wage burden and the machine burden in cases where a certain machine is operated by a man at definite wages, and cannot run without this man's attention.

Standards for operation are most important in factory management, and the correct cost-keeping system provides these standards.

If the fireman will not look at the steam gage, a danger signal may be provided, but the point is to make him look. If the manager of a factory installs a comprehensive cost-keeping system and fails to use the information which the system places at his command, he has only himself to blame if he does not take the trouble to look at the gage dial.

No. 1327

STANDARD CROSS-SECTIONS

By H. de B. Parsons, New York

Member of the Society

The advantage of a uniform method for cross-sectioning drawings so as to indicate graphically the materials, is so obvious that it needs no argument. For some years the tendency has been in this direction, with the result that many materials are represented on drawings by substantially uniform methods of cross-sectioning.

2 The author suggests that this Society promote this question of the use of uniform methods for indicating materials on engineering plans, and to this end supplements this paper with a sheet of typical standard cross-section designs (Fig. 1).

3 Communications were sent by the author to engineers, manufacturers, engineering societies and others having occasion to make cross-sectional drawings, in the United States, England, France and Germany, asking that they submit the standards, if any, used by them. The replies were given to the writer's assistant, Mr. David C. Johnson, who tabulated them and prepared the accompanying sheet of standard sections (Fig. 1), so that they might agree so far as possible with the standards now in use.

4 It is not practical to indicate every material by standard cross-sectioning, because there are so many alloys and materials of like nature which can always best be defined by lettering on the drawing.

5 The author believes that such a standard form as the one submitted would meet all requirements and create uniformity in practice which would be very desirable. Many have adopted standards of their own, but it would seem to be better if the same standards were used by all.

Presented at the Annual Meeting 1912, of The American Society of Mechanical Engineers.

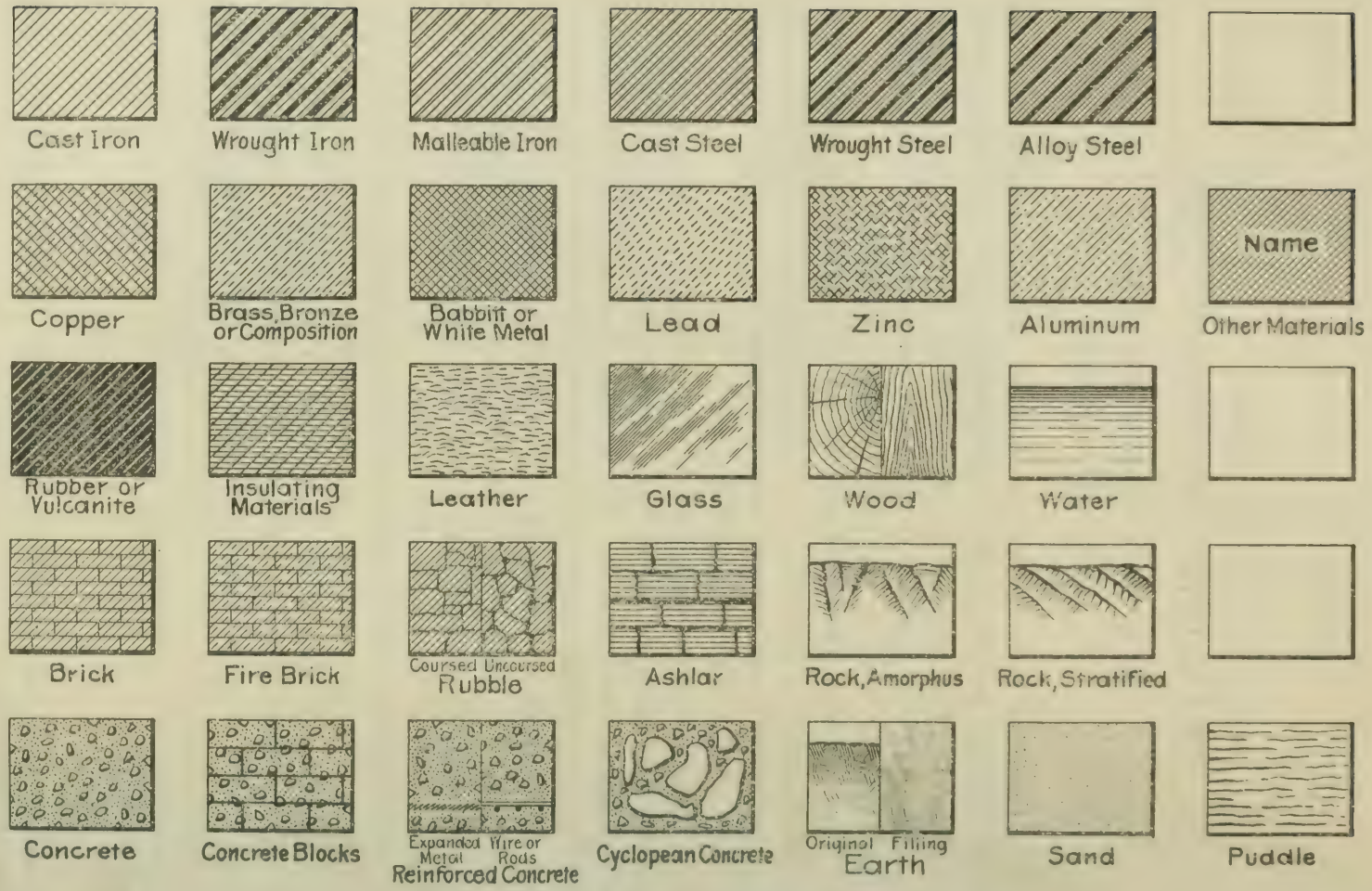


FIG. 1 TYPICAL STANDARD CROSS-SECTIONS

DISCUSSION

F. DeR. Furman. I would like to ask Mr. Parsons whether, in communicating with the different firms and engineers about the country, he got many replies indicating that tinting by pencil, carbon, or brush was used instead of cross-hatching for showing sections? I ask the question because several times during the past few years I have gone to different parts of the country to get information regarding drafting-room practice with a view to adopting a method of instruction for students in our drafting room. It seemed to me that there was considerable departure from the use of standard cross-sections, and that tinting, or smearing as it is sometimes called, was often used.

Another thing that I noted in connection with my inspection trips was that while, as it has been stated, different offices use different methods for representing the various materials, I found in several cities that many of the drafting rooms in that locality showed a certain conformity in their drafting-room standards. It was generally the case that each community had some leading old established manufacturing plant in which many of the present day independent manufacturers had had their early training, and these men had evidently carried with them into their new work, the drafting-room methods of their early experience. For example, in Providence I found that many of the concerns there used practically the same standards. So in bringing this matter to a uniform standard, it might not be a question so much of bringing together the methods of different manufacturers generally as of bringing together the methods employed by groups of manufacturers, thus making much easier the matter of standardization.

From the point of view of an instructor, I would like to see the Society adopt some standard convention for cross-sections.

The Author. In the replies to the inquiry sent out, there was uniformity up to a certain point. In Europe the prints are colored with water colors, generally using black and white prints. A tracing colored with crayon will not show on a blueprint or on a black and white print, but leaves the shaded part looking dirty. Smearing

with pencil makes a nice record on the blueprint, but distinctive marking can not be made by this method.

In reply to a question, the Navy has a standard for hatching and coloring, similar in many respects to those suggested by the author, although coloring is used only in exceptional cases and of late years has practically been abandoned.

If a standard method of indicating the material on cross-sections could be adopted universally, it would be a great assistance to all having occasion to read mechanical drawings. It would cause no additional labor as it is as easy to draw one style of sectioning as another.

No. 1328

TESTS OF LARGE BOILERS AT THE DETROIT EDISON COMPANY

By D. S. Jacobus, New York

Member of the Society

Many engineers have held the opinion that eventually much larger steam boilers will be used than those in service. The size of turbo-electric generating sets has increased with rapid strides and it might seem that an increase in the size of boilers would naturally follow. There has been a strong feeling, however, that it would be unwise to employ very large boiler units, as a failure of one of the large units would throw out of use a much greater proportion of power than the failure of a smaller unit.

2 Each of the boilers described in this paper is required, in the all-day practice of the power plant, to carry a load of 6000 kw. and in the evening to carry 7000 or 8000 kw. The experience in regular service of 18 months with the first boiler and of nine months with the second and third boilers has proven this to be good practice. The rated capacity of each boiler is 2365 h.p. on the basis of 10 sq. ft. of boiler heating surface per horsepower. Fourth and fifth boilers are now being erected and it is expected to complete the installation of ten boilers of this size within the next two years.

3 The engineering world is greatly indebted to the Detroit Edison Company and to Alex Dow, its general manager and a member of this Society, for pioneer work in installing boilers of two or three times the capacity of the largest boilers heretofore used, and for the opportunity of conducting the most elaborate and painstaking series of tests ever made on a boiler. The preliminary and regular tests required that the boiler room of a large power house be under the control of the observers for nearly three months, and for six weeks over 50 men worked in eight-hour shifts, night and day, exclusively on the tests. All the water and coal was accurately

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Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

weighed, and much work was done about the plant to avoid the possibility of leakage affecting the results. The magnitude of the undertaking may be appreciated when attention is drawn to the total quantities measured, which were as high as an average per hour of eight tons of coal, 75 tons of feedwater and one ton of ashes, the totals amounting to about 5000 tons of coal weighed and 45,000 tons of water.

4 The tests were run under the direct supervision of the writer, assisted by men chosen from the engineering staff of The Babcock & Wilcox Company. In addition to the men furnished by the Detroit Edison Company, The Solvay Process Company provided a number of observers. Their experimental engineer, Lewis C. Rogers, coöperated in securing the results and rendered most valuable assistance. Westinghouse, Church, Kerr & Company coöperated in designing apparatus and in making arrangements for the tests and was represented in the tests. The Solvay Process Company made heat determinations and analyses of the coal used in each test, and of the ashes. Duplicate samples were taken and the work was done a second time in the laboratory of The Babcock & Wilcox Company. The average of the results secured by the two laboratories was used in working up the tests. The tanks for weighing the water had each a capacity of 20,000 lb. Three such tanks and scales were provided, two being used regularly and the third held for a spare in case any irregularity developed in either of the other two. The coal was weighed by means of four special scales carrying overhead hoppers, each of about 2500 lb. capacity. The coal was conducted from the regular chutes into the hoppers, which were provided with hinged bottoms held in place by latches.

5 Two series of tests were made, one on a boiler fitted with Roney stokers, the other on a boiler fitted with Taylor stokers. Front and side views of the former are shown in Figs. 1 and 2. Fig. 3 shows a side view of the boiler fitted with Taylor stokers. Four Roney stokers were used in the furnace, two being at the front and two at the rear of the boiler. There was a low division wall between the stokers, and between the two sets of stokers a bridge wall. The Taylor stokers had 13 retorts on the front side and 13 on the rear, or 26 in all. The retorts on each side were set in a continuous row so as to provide an unbroken fire surface from one side of the boiler to the other. There was no bridge wall between the stokers, and when the dumping plates at the rear of each of the Taylor stokers were covered with fuel there was a continuous fuel bed beneath the entire boiler. The width of the furnace from one side to the other of the boiler was about 26½

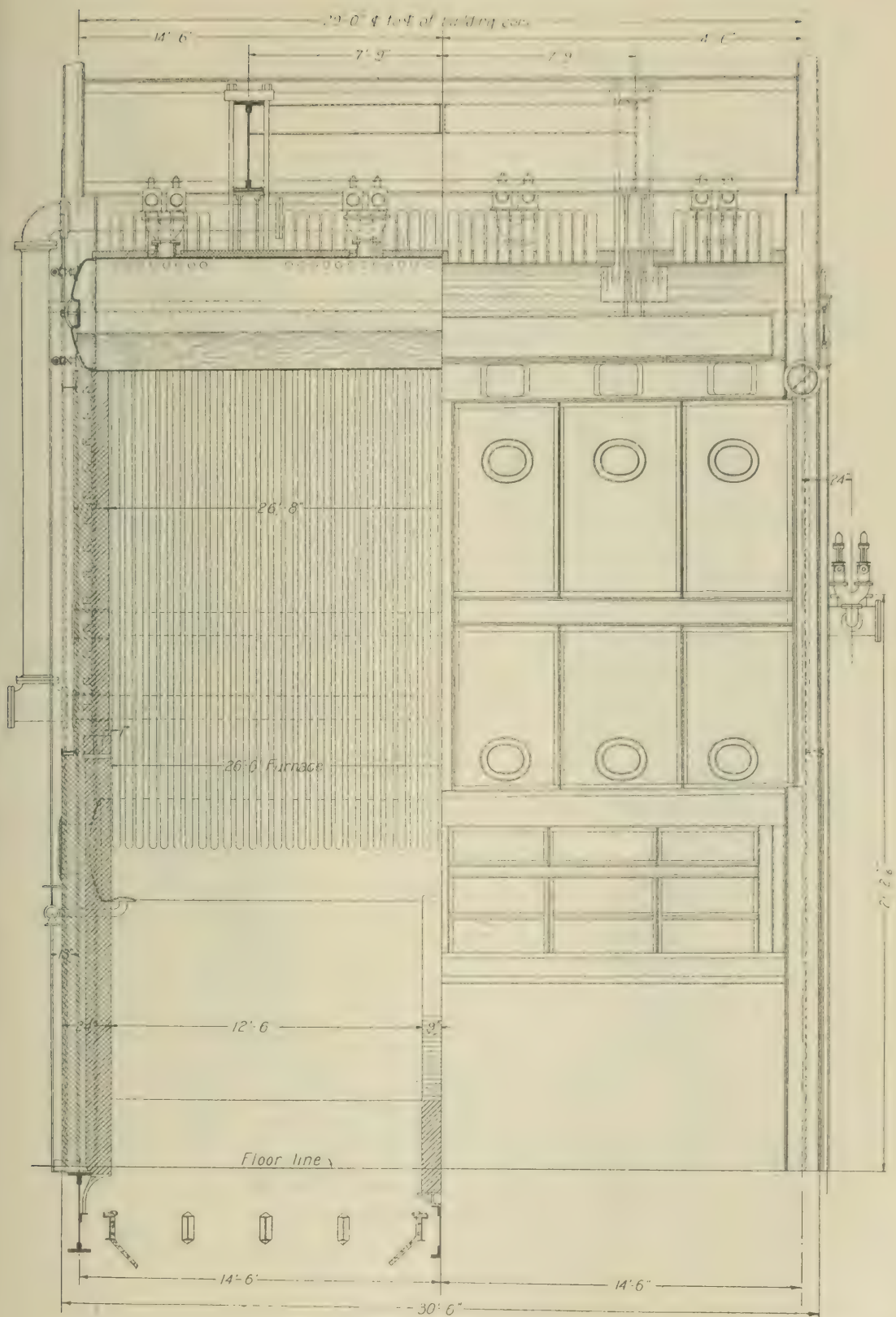


FIG. 1 FRONT AND HALF-SECTION OF BOILER OF 2365 RATED HORSEPOWER, FITTED WITH RONEY STOKER

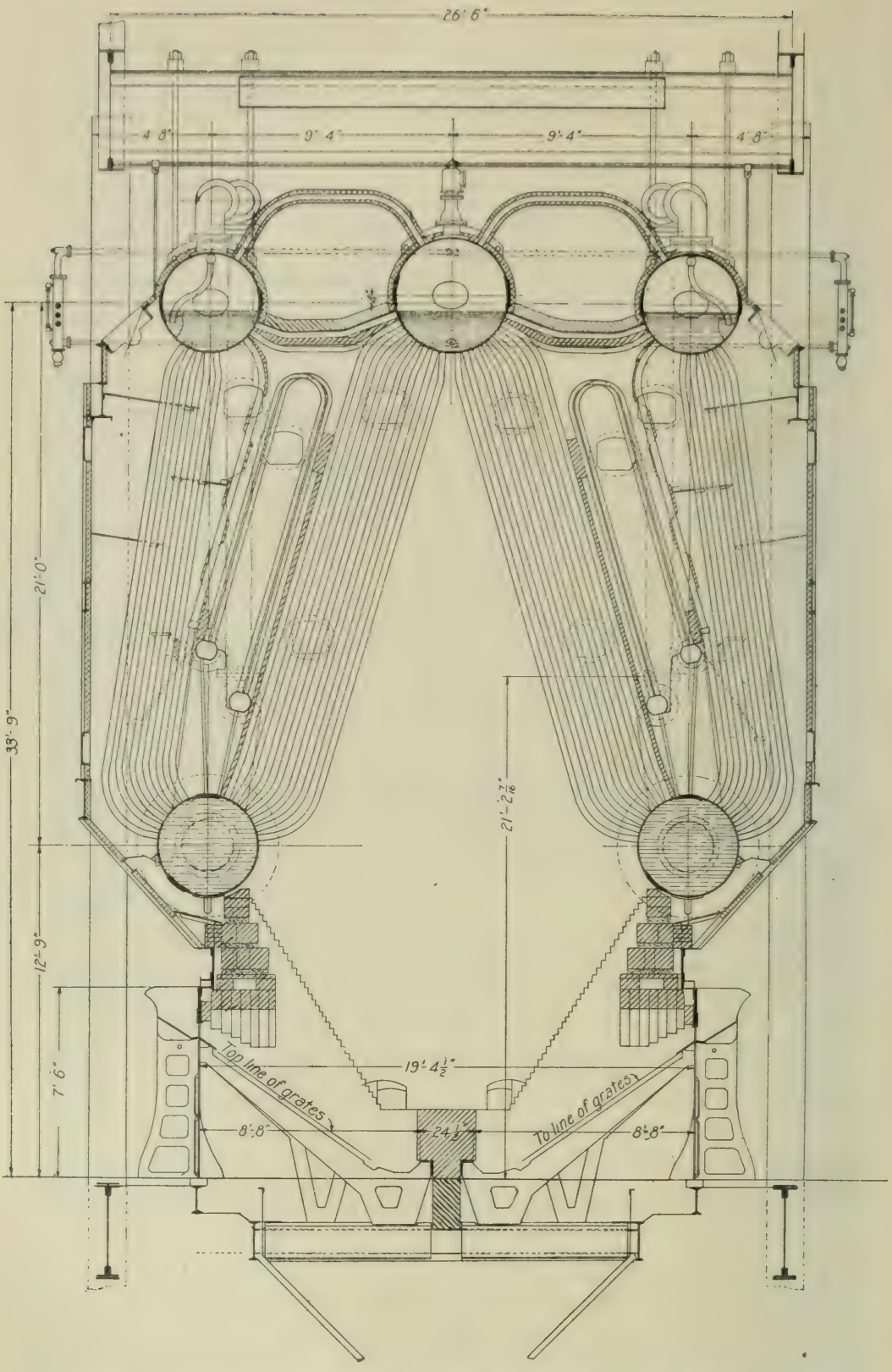


FIG. 2 LONGITUDINAL SECTION OF BOILER FITTED WITH RONEY STOKER

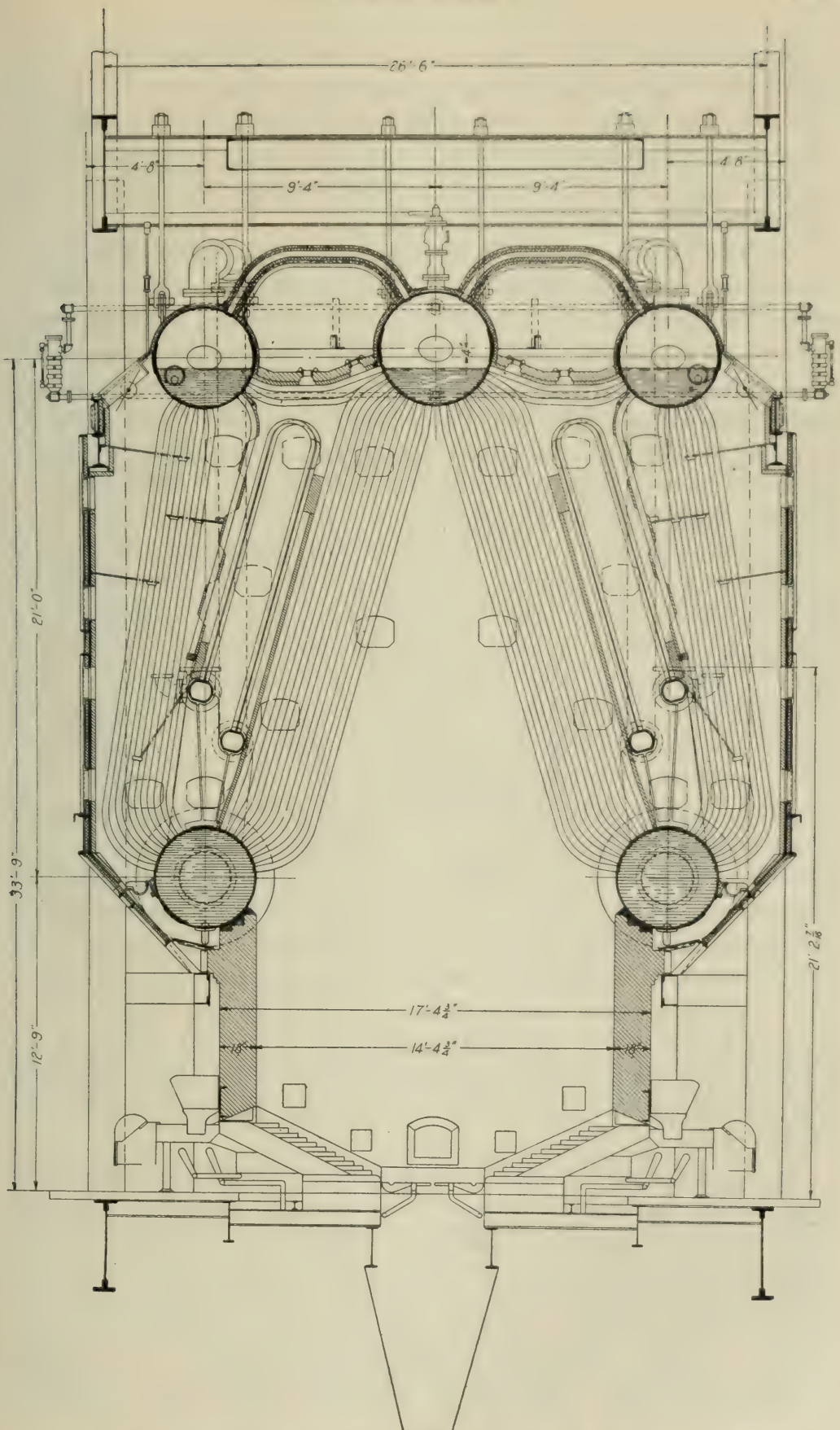


FIG. 3 LONGITUDINAL SECTION OF BOILER FITTED WITH TAYLOR STOKER

ft., and the depth of the furnace from the front to the rear of the boiler about 14 ft. The height of the furnace, measured above the dumping grates to the top of the first baffle, was about 29 ft. The large furnace volume combined with the particular form and height of the furnace had much to do with obtaining the high efficiencies. This will be treated more fully in Par. 13.

6 The boilers were built by The Babcock & Wilcox Company. Each boiler contained 23,654 sq. ft. of effective boiler heating surface and was provided with superheaters for supplying approximately 150 deg. of superheat. The grate surface, measured from the beginning to the rear of the dumping grates was 446 sq. ft. for the Roney stokers and 405 sq. ft. for the Taylor stokers. This method of measurement is not that usually employed in determining the grate surface of an underfeed stoker, but it was used in order to obtain comparative figures for the two stokers.

7 The design and installation of the plant and the general arrangement of the boilers and piping were under the charge of Westinghouse, Church, Kerr & Company. This firm designed the arrangement of the furnace brickwork for both the Taylor and Roney stokers; supervised the construction and followed up and corrected the minor defects which developed. Their engineering work in this connection involved the solution, on a large scale, of the problems which have always been troublesome to furnace designers, and it also involved the working out of special methods of construction demanded by the exceptionally high furnace temperatures. The ability of Westinghouse, Church, Kerr & Company to meet and dispose of the furnace and stoker problems which were anticipated and which arose was an important factor in determining both the makers and the users of boilers to undertake the new departure. Westinghouse, Church, Kerr & Company also coöperated in working out many of the construction details. This work and the general arrangement of the apparatus for the tests was under the charge of Henry O. Pond, to whom the author is indebted for valuable assistance.

8 In the development of the large units at Detroit, Mr. Dow expected an increase in economy due to minimizing the radiation losses, and to the high temperature obtainable in the large furnaces and combustion chamber. He also expected a reduction in the first cost, not on account of a reduced cost per unit of boiler surface, but to a less cost of the settings, floor space, suspension structures, valves and piping. Troubles through brickwork were anticipated and there were troubles which, in the main, have been overcome.

9 Another most important feature that had to be considered with so large a boiler unit was the possibility of tube troubles. In dealing with this point Mr. Dow inferred that their experience at the plant with the feedwater used in connection with boilers of a similar general type, warranted the conclusion that they should have practically no trouble with the tubes. This has been borne out by the experience with the boilers in actual service.

10 Mr. Dow credits his friend, Mr. W. H. Patchell, with the first idea of the possibilities of large boilers. The boilers and settings at Detroit differ greatly from those installed by Mr. Patchell where two boilers each of 10,850 sq. ft. of heating surface were mounted over a single furnace, but Mr. Dow says that to him is due the first con-

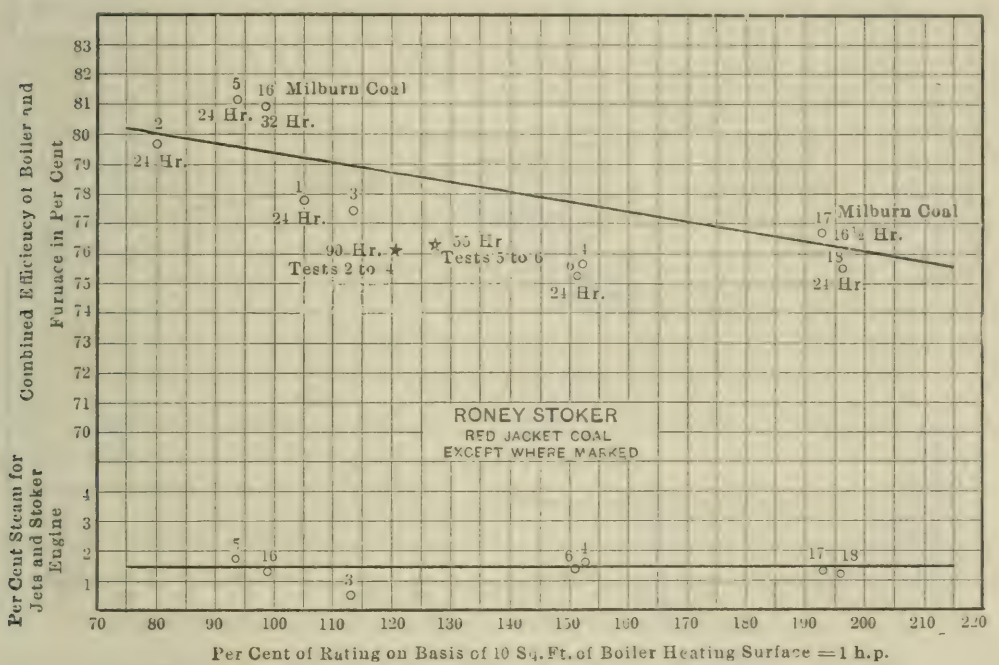

Combined Efficiency of Boiler and Furnace in Per Cent
Per Cent Steam for Jets and Stoker Engine
Per Cent of Rating on Basis of 10 Sq. Ft. of Boiler Heating Surface = 1 h.p.
16 Milburn Coal
24 Hr.
32 Hr.
24 Hr.
24 Hr.
90 Hr. ★ Tests 2 to 4
★ 55 Hr Tests 5 to 6
24 Hr.
17 Milburn Coal
16½ Hr.
24 Hr.
RONEY STOKER
RED JACKET COAL
EXCEPT WHERE MARKED

FIG. 4 RESULTS OF TESTS OF BOILER WITH RONEY STOKER

ception of the economical possibilities of the large units and the capacity obtainable through utilizing the whole floor area for the furnace.

11 A résumé of the results obtained is given in Tables 1 and 2 in the Appendix. Table 1 contains the results of tests of the boiler run with Roney stokers, and Table 2 of a boiler run with Taylor stokers. Figs. 4 and 5 show the results of these tests graphically.

12 It will be seen on examining the tables and figures that the combined efficiency of the boiler and furnace varies from about 80 per cent at slightly below rating, to about 76 per cent at double rating. In obtaining these efficiencies, the steam used for driving the stokers and for producing the forced blast for the Taylor stokers

has not been deducted from the total steam generated by the boiler. The amount of steam used by the Roney stokers was about $1\frac{1}{2}$ per cent of the total steam generated by the boiler, and for the Taylor stokers about $2\frac{1}{2}$ to 3 per cent. The effect that this steam would have on the plant economy depends on the ability to utilize the same for heating the feedwater. In a plant where the heat in the steam exhausted from the auxiliaries is returned to the feedwater there would be but little loss. In the case of the Taylor stokers all of the exhaust steam from the turbines driving the forced-blast fans may be carried to the feedwater heater, whereas with the Roney stokers only about $\frac{1}{3}$ of 1 per cent of the $1\frac{1}{2}$ per cent used may be so returned, since the

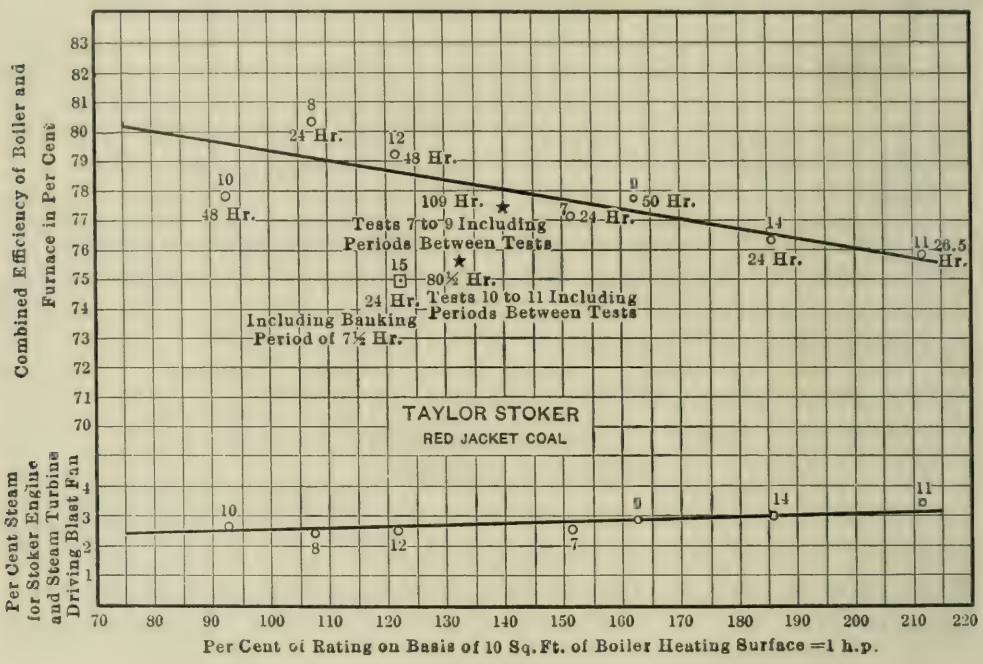

Combined Efficiency of Boiler and Furnace in Per Cent
Per Cent Steam for Stoker Engine and Steam Turbine Driving Blast Fan
Per Cent of Rating on Basis of 10 Sq. Ft. of Boiler Heating Surface =1 h.p.
24 Hr.
48 Hr.
109 Hr.
50 Hr.
24 Hr.
24 Hr.
80½ Hr.
24 Hr.
Tests 7 to 9 Including Periods Between Tests
Tests 10 to 11 Including Periods Between Tests
Including Banking Period of 7½ Hr.
TAYLOR STOKER
RED JACKET COAL

FIG. 5 RESULTS OF TESTS OF BOILER WITH TAYLOR STOKER

greater part of the steam is used in the jets which supply steam beneath the ignition arches of the stokers, and passes away with the chimney products. In a plant in which the auxiliaries are electrically driven, any power to operate the forced-blast apparatus would be a direct loss and, assuming in the case of the Taylor stoker that the steam consumption corresponding to the current required to drive the electrical auxiliaries is one-half of that found in the tests, the steam required to operate each of the stokers would be about the same. It therefore follows that the effect of the steam for driving the stokers on the station economy cannot be generalized, but that each plant must be considered by itself.

13 The efficiencies secured are exceptionally high, and the question may be asked whether this is due entirely to the large size of the boiler units. A careful study of the test data shows that the efficiencies were obtained with the most economical furnace conditions and that the thoroughness with which the fuel was consumed in the furnace had much to do with the attainment of the high figures. On examining Figs. 1, 2 and 3, showing the general arrangement of the boilers and settings, it will be seen that the roof of the furnace has an *A* shape, and that the coal is fired at the front and rear of the boiler setting. With this arrangement, the character of the combustion may be maintained constant from one side of the boiler to the other, and the loss so often experienced through a stream or lane of excess air or unconsumed combustible gases passing through the boiler and escaping to the stacks will be avoided. Those conversant with the subject know that in ordinary boiler furnaces there is a great variation in analyses taken from different points in the path of the flue gases leaving the furnace, whereas with the present furnace arrangement this action is reduced to a minimum, as any irregularity from the front to the rear of the grates disappears before the gases pass from the upper part of the combustion chamber, and as the composition of the gases for uniform firing conditions will be uniform from one side of the boiler to the other, or from right to left in Fig. 1, it follows that the composition of the entire volume of the gases leaving the furnace will be substantially uniform.

14 A straight reference line is drawn in Figs. 4 and 5 in order that the results obtained by the two stokers may be readily compared, this line being the same on both diagrams. It will be noted that the test results fall on both sides of the line, and that the efficiency obtainable with each stoker is about the same. Naturally a curve should be used instead of a straight line to show the variation in the efficiency, but the accidental variation in the test results is such that it would be impossible to trace an exact curve.

15 On examining Fig. 4, it will be seen that the efficiencies, where several tests are worked out together, including the periods between the tests, are only about $1\frac{1}{2}$ per cent lower than would be indicated by averaging the results for the separate tests by passing a straight line between the points. The same follows for the tests which are worked up together for the Taylor stoker. This shows that including the periods between the tests, during which the dust was blown from the exterior of the tubes and the firing was not watched as carefully as it was during the tests, did not greatly lower the efficiency.

It was proposed at first to run a test of a week or more on each of the stokers, but this plan was abandoned, since there was trouble through leakage in the feed pump which made it necessary to shut down, and on reconsidering the matter it was deemed advisable to give up running as long a test as this since it developed that a week would be too long a period to run without blowing down the boilers. Test No. 15, with the Taylor stoker, included a banking period of $7\frac{1}{2}$ hours. It will be noted that the efficiency including the banking period is about 75 per cent. The capacity of the boiler for each hour of this test is shown in Fig. 6.

16 It will be seen (Figs. 4 and 5) that the average amount of steam used for driving the Roney stoker was about $1\frac{1}{2}$ per cent at all loads, and that it varied from about $2\frac{1}{2}$ per cent at rating to about 3 per cent double rating for the Taylor stoker. Saturated steam

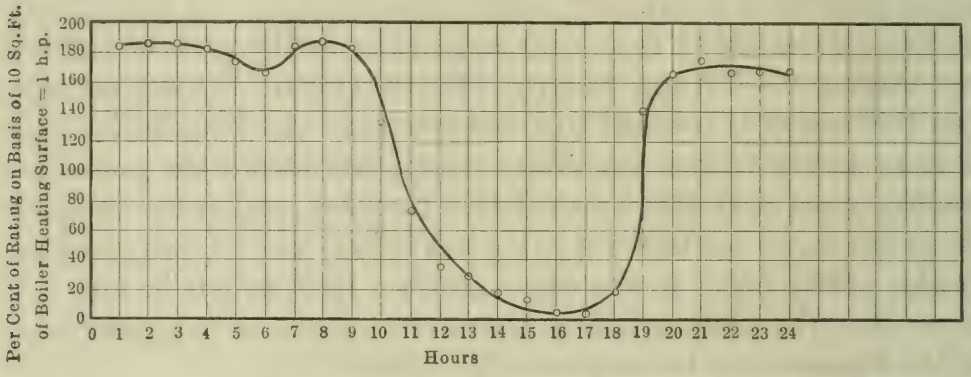


FIG. 6 CAPACITY OF BOILER WITH TAYLOR STOKER FOR EACH HOUR OF A TEST WHERE THE FIRES WERE BANKED FOR $7\frac{1}{2}$ HOURS

from another boiler was used for driving the auxiliaries. The saturated steam was reduced to its equivalent weight of superheated steam in determining the percentages above given.

17 The minimum length of a regular test was 24 hours. The efficiency for a test of $16\frac{1}{2}$ hours is plotted in one diagram. In this particular test a special coal was used which ran out and thus brought the test to a close. Some of the tests were of 48 hours' duration and over. A continuous record was kept and, as stated in Par. 15, the efficiency was determined for periods embracing several of the tests including the intervals between them. Intervals were selected from some of the tests and were worked up to obtain comparative results. Figures derived in this way are given in Table 3, and it will be noted that there is a substantial agreement between the results secured in the regular tests and the results for the selected intervals. It must not be inferred from this that sufficient accuracy would

be secured by shortening the length of the tests. The tests reported in this paper were made after a painstaking series of preliminary tests with both of the stokers, and the operators were well trained in maintaining uniform conditions. Twenty-four hours is none too long an interval for obtaining accurate results with an underfeed stoker of the class tested, as one can readily appreciate when his attention is called to the fact that the generation of steam will continue after no more coal is fed to the furnace with but little difference in the appearance of the fire. There is no method for measuring the fire except sizing it up with the eye, as one must form an estimate of the amount of clinker present. One of the operators at the plant was earnest in his opinion that the fires as balanced up by the author for the Taylor stoker contained from $2\frac{1}{2}$ to 5 tons more coal at the end of a test than at the beginning, a contention that was shown beyond doubt to be in error by the heat balances. This goes to show how far the estimates of the amount of coal may affect the results and that consequently there is great danger of obtaining erroneous figures in short tests. Where the fire may be kept in a comparatively uniform condition, as in the Roney stoker, it might seem that the tests could be shortened, but even in such cases the author is in favor of making 24-hour tests.

18 Great care was taken to prevent the possibility of leakage affecting the tests. All fittings where there might be leakage were either blanked off or two valves were provided with an open drip between them. The boilers were tested with hydraulic pressure both before and after the tests. Special leakage tests were also made in which the hot boilers were completely filled with water and the full pressure maintained on the boilers and feed mains, and in each case there was no leakage indicated in tests of several hours' duration.

19 Tables 4 and 5 give the average analyses of the flue gases. It will be seen that the furnace efficiency was exceptionally high for both of the stokers.

20 Tables 6 and 7 give the analyses of the coal and the heats of combustion as determined in the laboratories of The Solvay Process Company and The Babcock & Wilcox Company, as well as the average figures used in computing the results of the tests.

21 Table 8 gives a comparison of the results secured for the heat of combustion of the coal as determined in The Solvay Process Company laboratory and in the laboratory of The Babcock & Wilcox Company. In this table the values are worked out per pound of combustible, the percentage of ash in each case being that determined

by the individual laboratory. The heat of combustion per pound of combustible, computed from the results of the ultimate analyses made by the two laboratories, is given, and it may be seen that the figures obtained by the oxygen-bomb calorimeter correspond very closely with the computed ones. The coal samples were obtained by taking a small amount of coal each time it was weighed and plac-

FIG. 7 SPECIAL APPARATUS FOR WEIGHING FEEDWATER

ing it in a covered air-tight receptacle. Smaller samples for the two laboratories were obtained independently from the large sample of coal collected during the test and the results of the two laboratories were therefore influenced by any variation in the smaller samples.

22 Tables 9 to 11, inclusive, contain the average test data and computed results for the tests with the Roney stoker, and Tables

12 to 14, inclusive, the average test data and computed results for the tests with the Taylor stoker.

23 Tables 15 and 16 give the results of heat balances. Two sets of heat balances were made, one applying to the gases after they left the boiler dampers, and the other to the gases at the top of the second pass directly beneath the boiler dampers. The analyses of the gases after leaving the boiler dampers were made with an Orsat apparatus, the samples of gas being obtained from the middle points

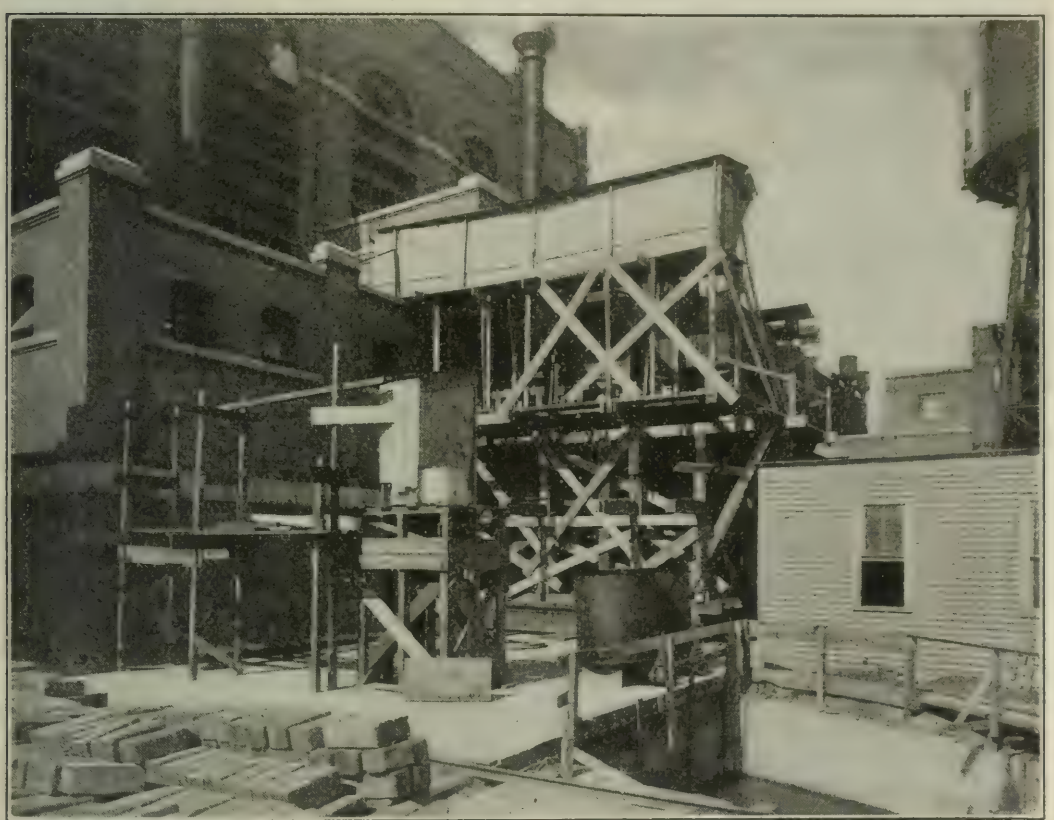


FIG. 8 SPECIAL APPARATUS FOR WEIGHING FEEDWATER

of the flues. The temperature of the gases directly beneath the boiler dampers at the top of the second pass was determined by means of electrical couples, two sets being used at each side of the boiler. The analyses of the flue gases at this point were made by a Hempel apparatus, an average sample being obtained by drawing the gas from six points at the front and six points at the rear of the boiler. The gas was drawn through the collecting piping with an aspirator, and a sample of the mixed gases was drawn into a collecting bottle for

analysis. In order to make sure that the same weight of gas entered each of the six sampling tubes, a throttling cock was adjusted to make the suction on each tube a given amount, as indicated by a water gage. The Hempel apparatus was also used in the way just described for analyzing the gases at the bottom of the second pass. The apparatus for obtaining the average samples was constructed by The Solvay Process Company.

FIG. 9 SPECIAL HOPPERS AND COAL-WEIGHING SCALES

24 The results of the heat balances show that the average radiation and unaccounted-for losses are only about $2\frac{1}{2}$ to 3 per cent. There is naturally a variation one way or the other due to accidental errors, but the results are as uniform as could be expected in work of the sort.

25 Preliminary tests were first made on the boiler with the Taylor stoker. The apparatus was then shifted and preliminary tests

were made on the boiler with the Roney stoker. A careful study of the operating conditions for the best economy was made in the preliminary tests. The results of the regular tests only are given in this paper.

26 Readings of temperatures, pressures, etc., were taken every half hour. The water was balanced each hour. The coal weighed per hour and supplied to the hoppers was balanced each hour and recorded, but no attempt was made to obtain a correct coal balance

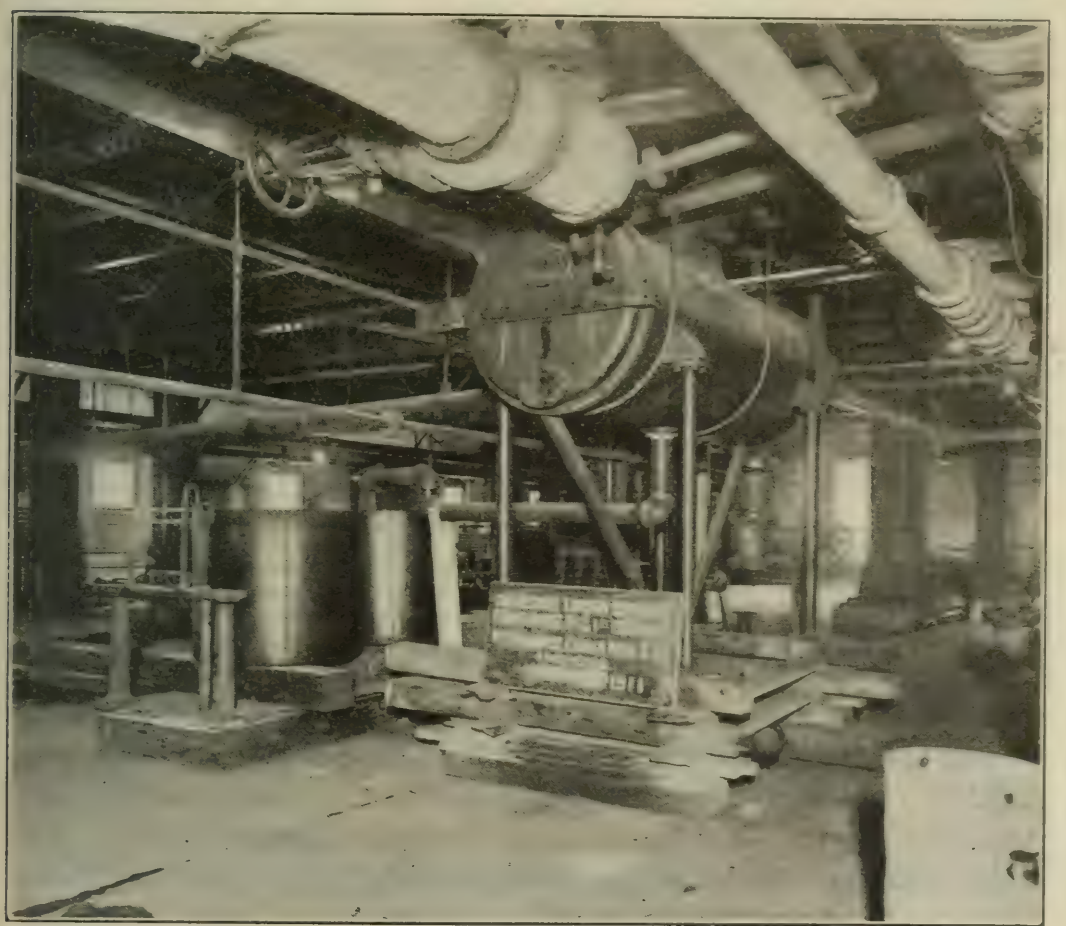


FIG. 10 SPECIAL CONDENSER AND WEIGHING APPARATUS FOR DETERMINING QUANTITY OF STEAM USED BY STOKER AUXILIARIES

except at the beginning and ending of the tests, as this necessitated running the coal quite low in the hoppers of the stokers. As soon as the data were taken, a copy was made on a large sheet in the boiler room, so that those conducting the tests could, at a glance, note whether the conditions were uniform. Marks and Davis steam tables were used in working up the results.

27 Two views of the special apparatus erected for weighing the feedwater are shown in Figs. 7 and 8. Water from the heater was elevated by means of a centrifugal pump specifically provided for the purpose and discharged into tanks supported on special platform scales. Two of these tanks appear in Fig. 8, the third being in the shadow of the building. The men weighing the water were protected from the weather by a rough housing placed above the level of the tanks, the scale beams being inside this housing. The water from the tanks ran into two lower horizontal tanks, one of which is plainly visible in Fig. 7. The lower tanks were provided with gage glasses and the water was brought to an accurate level every hour. The water from the lower tanks was pumped directly into the boilers by one of the regular centrifugal feed pumps used at the plant. Any leakage in the glands of the pump was caught and

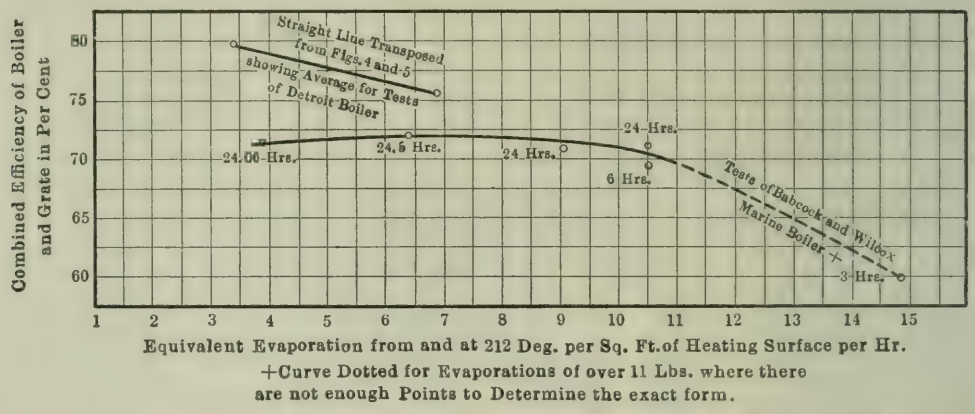

Combined Efficiency of Boiler and Grate in Per Cent
80
75
70
65
60
Straight Line Transposed from Figs. 4 and 5 showing Average for Tests of Detroit Boiler
24 Hrs.
24.00 Hrs.
24.5 Hrs.
24 Hrs.
6 Hrs.
Tests of Babcock and Wilcox Marine Boiler +
3 Hrs.
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15
Equivalent Evaporation from and at 212 Deg. per Sq. Ft. of Heating Surface per Hr.
+Curve Dotted for Evaporations of over 11 Lbs. where there are not enough Points to Determine the exact form.

FIG. 11 COMPARISON OF RESULTS WITH THOSE FROM A B. & W. MARINE BOILER

returned to the large horizontal tanks by a small centrifugal pump. Two of the special hoppers and a portion of the coal-weighing scales are shown in Fig. 9. Fig. 10 shows a special condenser and weighing apparatus erected for determining the amount of steam used by the stoker auxiliaries. All scales were accurately calibrated to their full capacity before and after the tests of each of the boilers by means of standard weights.

28 In order to compare the results with others where high efficiencies have been secured, Fig. 11 is presented. In this figure the straight line shown in Figs. 4 and 5 is transferred and marked, Average for Tests of Detroit Boiler. The plotted results for individual tests shown in Fig. 7, are for tests made by a Board of United States Naval Officers on a hand-fired Babcock & Wilcox marine boiler and reported in the Journal of the American Society of Naval Engineers, November 1910. By combining the results secured with the two

boilers it will be seen that the efficiency varies from about 80 per cent at an evaporation of 3 lb. per sq. ft. of heating surface per hour from and at 212 deg. fahr. to 76 per cent at 7 lb., 72 per cent at 10 lb. and 60 per cent at 14.7 lb. It therefore follows that if the performance of the two boilers could be combined, a boiler could be run from about 80 per cent of the ordinary rating to over four times this rating, and for most classes of central power-plant service it would be possible to run all of the boilers in the plant all of the time, thus eliminating the loss occasioned through having to carry a number of boilers under banked fires. The writer is now working on the development of a boiler of this sort.

APPENDIX

TABLES GIVING DATA AND RESULTS OF TESTS

TABLE 1 TESTS WITH RONEY STOKER. RÉSUMÉ OF PRINCIPAL RESULTS

No. of Test	Length, Hr.	Per Cent Rating	B.t.u. in Coal	Per Cent Ash in Dry Coal	Efficiency	Per Cent Steam used by Stoker Engines and Steam Jets	Per Cent Combustible in Ash	Temp. of Flue Gases Leaving Boiler, Deg. Fahr.
1	25	105.0	14362	5.98	77.84	....	19.6	576
2	24	80.0	14225	6.52	79.88	....	17.9	480
3	24	113.8	14308	7.40	77.45	0.63	24.4	542
4	30	152.4	13756	6.54	75.78	1.58	30.8	670
5	24	94.0	13896	6.89	81.15	1.75	31.6	483
6	24	150.7	14037	6.13	75.28	1.45	26.7	662
16	32	98.6	14476	9.30	80.98	1.34	34.1	460
17	16.5	193.3	14493	8.24	76.73	1.39	24.6	636
18	24	195.7	13689	9.81	75.57	1.32	23.2	694
2–4†	90	119.8	14098	6.81	76.13	....	25.8	572
5–6†	55	127.3	13977	6.84	76.23	....	29.4	575

† Including periods between tests.

TABLE 2 TESTS WITH TAYLOR STOKER. RÉSUMÉ OF PRINCIPAL RESULTS

No. of Test	Length, Hr.	Per Cent Rating	B.t.u. in Coal	Per Cent Ash in Dry Coal	Efficiency	Per Cent Steam used by Stoker Auxiliaries*	Per Cent Combustible in Ash	Temp. of Flue Gases Leaving Boiler, Deg. Fahr.
7	24	151.2	14000	7.03	77.07	2.61	31.5	575
8	24	107.9	13965	6.34	80.28	2.44	27.1	493
9	50	162.8	13998	6.75	77.85	2.87	31.3	574
10	48	92.9	14188	9.90	77.90	2.63	27.2	487
11	26.5	211.3	14061	9.55	75.84	3.41	36.1	651
12	48	121.3	14010	8.09	79.24	2.57	27.6	535
14	24	185.5	14272	8.71	76.42	2.95	28.8	647
15†	24	123.1	14213	8.34	74.90	2.77	30.5	561
7– 9‡	109	140.0	13983	7.22	77.66	2.68	30.0	545
10–11‡	80.5	132.8	14095	9.58	75.66	3.04	31.1	542

* Engines driving stokers and steam turbine driving fan.

† In test No. 15 the fires were banked for 7½ hr. and the averages include this period.

‡ Including periods between tests.

TABLE 3 TESTS WITH TAYLOR STOKER

RESULTS OF REGULAR TESTS COMPARED WITH RESULTS FOR INTERVALS, SELECTED FROM THE REGULAR TESTS, AND RESULTS FOR A SHORT TEST

Test No.	Date of Trial and Time	Length, Hr.	Per Cent Rating	B.t.u. in Coal	Per Cent Ash in Dry Coal	Efficiency	Per Cent Steam used by Stoker Auxiliaries	Per Cent Combustible in Ash	Temp. of Flue Gases leaving Boiler, Deg. Fahr.
10*A*	6–8, 1 p.m. to 6–9, 1 p.m.	24	94.4	14250	9.84	77.63	2.64	26.9	489
10	6–7, 4 p.m. to 6–9, 4 p.m.	48	92.9	14188	9.90	77.90	2.63	27.2	487
11*A*	6–10, 10.30 a.m. to 10.30 p.m.	12	214.8	14049	10.48	76.18	3.66	36.1	668
11	6–9, 10 p.m. to 6–11, 12.30 a.m.	26.5	211.3	14061	9.55	75.84	3.41	36.1	651
12*A*	6–12, 3 p.m. to 6–13, 3 p.m.	24	117.0	14027	7.73	79.21	2.52	29.7	514
12*B*	6–13, 3 p.m. to 6–14, 3 p.m.	24	125.5	13994	8.45	79.19	2.62	25.4	555
12	6–12, 3 p.m. to 6–14, 3 p.m.	48	121.3	14010	8.09	79.24	2.57	27.6	535
13	6–14, 9 p.m. to 6–15, 8.30 a.m.	11.5	203.2	14056	8.11	76.6†	3.29	35.3	641

† Not shown in Fig. 5 as test is too short to be relied on.

TABLE 4 FLUE GAS ANALYSES—RONEY STOKER

No. of Test	Bottom of Last Pass			Top of Last Pass			In Flue		
	CO_2	O	CO	CO_2	O	CO	CO_2	O	CO
1	13.22	5.29	0.00	12.41	6.48	0.00	11.95	7.55	0.05
2	15.18	3.00	0.06	14.31	4.01	0.07	14.33	4.54	0.11
3	14.50	3.50	0.09	12.25	6.12	0.02	13.05	6.46	0.18
4	14.45	3.44	0.35	13.51	4.68	0.20	14.74	3.96	0.54
5	15.65	2.27	0.25	14.68	3.40	0.20	14.40	4.54	0.35
6	14.77	3.23	0.20	14.28	3.87	0.15	14.66	4.23	0.31
16	13.82	4.88	0.00	13.82	4.88	0.00	13.55	5.92	0.07
17	14.25	4.06	0.40	13.98	4.48	0.25	14.69	4.55	0.20
18	.....	....	....	.....	....	....	14.16	5.04	0.16

TABLE 5 FLUE GAS ANALYSES—TAYLOR STOKER

No. of Test	Bottom of Last Pass			Top of Last Pass			In Flue		
	CO_2	O	CO	CO_2	O	CO	CO_2	O	CO
7	15.46	2.83	0.08	12.16	6.64	0.00	14.00	5.50	0.42
8	15.04	3.35	0.02	12.74	5.83	0.01	13.69	5.82	0.10
9	15.84	2.40	0.03	13.88	4.62	0.02	14.74	4.57	0.12
10*A*	13.40	5.38	0.00	12.09	6.79	0.00	12.05	7.74	0.06
10	13.14	5.59	0.00	11.91	6.96	0.00	11.86	7.96	0.06
11*A*	16.50	1.55	0.24	15.18	3.01	0.17	15.39	3.98	0.13
11	15.25	2.92	0.25	14.62	3.30	0.21	15.45	3.86	0.17
12*A*	14.85	3.61	0.00	13.30	5.30	0.00	13.73	5.81	0.04
12*B*	14.80	3.59	0.00	13.27	5.41	0.00	13.86	5.64	0.04
12	14.83	3.59	0.00	13.28	5.35	0.00	13.79	5.73	0.04
13	15.43	2.96	0.09	13.85	4.65	0.37	15.17	3.90	0.19
14	15 07	3.33	0.17	12.90	5.67	0.06	14.20	5.08	0.08
15	11.70	7.16	0.12	10.35	8.79	0.01	10.83	8.93	0.09

TABLE 6 RONEY STOKER. COAL AND ASH ANALYSES, PERCENTAGES BY WEIGHT

No. of Test		Proximate Analysis of Coal				B.t.u. in Coal	Combustible in Ash	Moisture in Ash	Ultimate Analysis of Dry Coal					
		Fixed Carbon	Volatile Matter	Moisture	Ash				C	H	O	N	S	Ash
1*	S	59.50	35.24	1.82	5.26	14369	18.04	23.37	80.35	5.22	7.03	0.96	0.88	5.56
	B	63.69	31.15	2.17	5.16	14354	21.27	24.39	78.67	6.15	7.76	1.20	1.06	5.16
	A	61.59	33.19	2.00	5.22	14362	19.65	23.88	79.51	5.69	7.39	1.08	0.97	5.36
2	S	59.44	34.08	1.30	6.48	14225	15.56	24.37	79.45	5.15	6.36	1.06	1.08	6.90
	B	63.55	30.31	2.36	6.14	14224	20.23	23.93	77.69	6.04	7.67	1.17	1.29	6.14
	A	61.49	32.19	1.83	6.32	14225	17.90	24.15	78.57	5.60	7.02	1.11	1.18	6.52
3	S	59.31	34.95	1.58	5.74	14400	29.72	21.96	80.73	5.30	5.56	1.06	1.00	6.35
	B	61.01	32.78	1.83	6.21	14215	19.18	20.56	76.80	6.10	8.61	1.19	1.09	6.21
	A	60.16	33.86	1.71	5.98	14308	24.45	21.26	78.76	5.70	7.09	1.13	1.04	6.28
4	S	58.01	33.98	1.94	8.01	13529	32.84	22.44	77.21	4.89	7.02	1.15	0.94	8.79
	B	62.89	31.13	1.66	5.98	13984	28.76	23.46	76.42	5.67	9.80	1.04	1.09	5.98
	A	60.45	32.56	1.80	6.99	13756	30.80	22.95	76.81	5.28	8.41	1.10	1.02	7.38
5	S	58.56	34.83	1.52	6.61	13747	28.63	23.23	79.14	4.72	6.51	1.13	1.42	7.08
	B	62.73	32.23	2.08	5.04	14045	34.50	26.86	74.74	5.98	9.92	1.09	1.34	6.93
	A	60.65	33.53	1.80	5.82	13896	31.57	25.05	76.94	5.35	8.22	1.11	1.38	7.00
6	S	60.02	33.40	2.35	6.58	13572	24.94	23.04	79.14	4.85	6.65	1.15	1.18	7.03
	B	62.17	33.23	2.09	4.60	14501	28.54	23.65	79.68	6.34	7.62	1.22	1.11	4.03
	A	61.10	33.31	2.22	5.59	14037	26.74	23.34	79.41	5.60	7.14	1.18	1.14	5.53
16	S	63.57	27.81	2.78	8.62	14155	38.82	15.79	79.95	5.31	4.36	1.25	0.64	8.49
	B	66.94	27.84	2.47	5.22	14797	29.33	.....	78.86	5.72	8.36	1.04	0.80	5.22
	A	65.25	27.83	2.62	6.92	14476	34.08	15.79	79.41	5.51	6.36	1.15	0.72	6.85
17	S	63.29	28.76	2.28	7.95	14265	28.25	16.67	80.98	5.37	3.49	1.27	0.61	8.28
	B	66.49	27.70	2.62	5.81	14721	20.96	12.08	81.25	5.65	5.27	1.10	0.92	5.81
	A	64.89	28.23	2.45	6.88	14493	24.61	14.38	81.12	5.51	4.38	1.18	0.77	7.04
18	S	57.36	31.63	2.06	11.01	13385	25.29	17.33	75.74	5.09	5.27	1.12	1.17	11.61
	B	62.30	29.75	2.28	7.95	13993	21.08	14.68	77.44	5.43	7.18	0.99	1.01	7.95
	A	59.83	30.69	2.17	9.48	13689	23.19	16.01	76.59	5.26	6.22	1.06	1.09	9.78
2-4†	S	58.81	34.38	1.61	6.81	14042	27.04	22.80	.....	.....	.....	.....	.....	.....
	B	62.18	31.53	1.90	6.29	14155	24.54	22.92	.....	.....	.....	.....	.....	.....
	A	60.50	32.95	1.75	6.55	14098	25.79	22.86	.....	.....	.....	.....	.....	.....
5-6†	S	59.14	34.22	1.95	6.64	13677	27.32	23.18	.....	.....	.....	.....	.....	.....
	B	62.35	32.77	2.08	4.88	14277	31.52	23.00	.....	.....	.....	.....	.....	.....
	A	60.74	33.50	2.01	5.76	13977	29.42	23.09	.....	.....	.....	.....	.....	.....

* *S*, Determined by The Solvay Process Co.; *B*, Determined by The Babcock & Wilcox Co.; *A*, Average used in working up results.

† Includes periods between tests and figures are based on analyses of all of the coal used.

TABLE 7 TAYLOR STOKER. COAL AND ASH ANALYSES, PERCENTAGES BY WEIGHT

No. of Test		Proximate Analysis of Coal				B t.u. in Coal	Combustible in Ash	Moisture in Ash	Ultimate Analysis of Dry Coal					
		Fixed Carbon	Volatile Matter	Moisture	Ash				C	H	O	N	S	Ash
7*	S	60.27	34.88	1.70	4.85	14240	31.54	22.12	79.90	5.06	7.71	1.16	1.03	5.14
	B	62.09	31.35	2.13	6.56	13759	.....	.....	76.95	6.06	8.79	1.02	0.97	6.21
	A	61.18	33.11	1.91	5.71	14000	31.54	22.12	78.42	5.56	8.25	1.09	1.00	5.68
8	S	59.63	33.76	1.78	6.61	14009	27.07	49.36	78.91	5.13	6.87	1.05	1.15	6.89
	B	63.64	31.15	2.04	5.21	13920	.....	.....	75.48	5.52	11.24	1.18	1.37	5.21
	A	61.63	32.46	1.91	5.91	13965	27.07	49.36	77.19	5.33	9.05	1.12	1.26	6.05
9	S	58.34	33.29	2.12	8.37	13769	35.46	40.79	78.25	4.80	6.33	0.99	1.22	8.41
	B	61.25	32.63	2.13	6.12	14227	27.16	32.00	77.04	5.64	9.14	1.12	0.93	6.13
	A	59.80	32.96	2.12	7.24	13998	31.31	36.40	77 64	5.22	7.74	1.05	1.08	7.27
10A	S	59.95	33.54	1.71	6.51	13892	28.83	18.61	.....					
	B	60.32	33.92	2.10	5.76	14607	24.90	20.75	.....					
	A	60.14	33.73	1.90	6.13	14250	26.87	19.68	.....					
10	S	58.98	34.17	1.92	6.85	13876	29.07	19.51	79.28	5.14	6.27	1.19	1.11	7.01
	B	60.26	33.97	2.20	5.77	14500	25.42	20.42	78.76	5.85	7.57	1.13	0.92	5.77
	A	59.62	34.07	2.06	6.31	14188	27.25	19.97	79.02	5.50	6.92	1.16	1.01	6.39
11A	S	57.79	35.39	1.76	6.82	13862	37.63	18.58	.....					
	B	60.32	33.75	2.04	5.93	14235	34.62	18.76	.....					
	A	59.06	34.57	1.90	6.37	14049	36.13	18.67	.....					
11	S	58.80	34.49	1.70	6.71	13889	37.63	18.58	79.89	4.86	6.36	1.18	0.90	6.81
	B	62.36	32.47	2.10	5.17	14232	34.62	18.76	77.44	5.80	9.30	1.13	1.06	5.27
	A	60.58	33.48	1.90	5.94	14061	36.13	18.67	78.66	5.33	7.83	1.16	0.98	6.04
12A	S	57.59	33.38	1.90	9.03	13585	31.62	15.77	.....					
	B	62.10	32.32	2.07	5.58	14469	27.76	28.20	.....					
	A	59.84	32.85	1.98	7.31	14027	29.69	21.99	.....					
12B	S	60.05	33.74	1.76	6.21	13996	28.26	15.43	.....					
	B	63.62	31.37	2.06	5.01	13991	22.54	16.16	.....					
	A	61.84	32.55	1.91	5.61	13994	25.40	15.80	.....					
12	S	58.82	33.56	1.83	7.62	13790	29.94	15.60	78.89	5.10	5.83	1.30	1.13	7.75
	B	62.86	31.84	2.06	5.30	14230	25.15	22.18	76.00	5.81	10.79	1.06	1.05	5.29
	A	60.84	32.70	1.95	6.46	14010	27.55	18.90	77.44	5.46	8.31	1.18	1.09	6.52
13	S	60.18	33.16	1.79	6.66	14141	41.61	16.00	.....					
	B	62.89	31.92	2.06	5.19	13970	29.10	16.30	.....					
	A	61.54	32.54	1.92	5.92	14056	35.35	16.15	.....					
14	S	59.82	34.34	2.00	5.84	14111	32.75	13.70	80.73	5.10	6.03	1.27	1.01	5.86
	B	63.24	32.09	1.80	4.67	14433	24.92	14.71	76.80	5.71	10.14	1.02	1.01	5.32
	A	61.53	33.21	1.90	5.26	14272	28.84	14.21	78.77	5.41	8.08	1.14	1.01	5.59
15	S	61.30	33.24	1.97	5.46	14231	32.27	23.08	.....					
	B	62.99	31.75	2.38	5.26	14194	28.77	19.88	.....					
	A	62.14	32.50	2.18	5.36	14213	30.52	21.48	.....					
7-9†	S	59.18	33.65	1.95	7.17	13900	32.12	36.72	.....					
	B	61.70	32.19	2.11	6.11	14067	27.82	29.11	.....					
	A	60.44	32.92	2.03	6.64	13983	29.97	32.91	.....					
10-11†	S	58.82	34.12	1.90	7.06	13830	32.65	19.17	.....					
	B	61.07	33.19	2.20	5.74	14361	29.49	19.92	.....					
	A	59.55	33.65	2.05	6.40	14095	31.07	19.54	.....					

*S Determined by The Solvay Process Co.; B, Determined by The Babcock & Wilcox Co.; A, Average used in working up results.

† Includes periods between tests and figures are based on the analyses of all of the coal used.

TABLE 8 COMPARISON OF THE HEATS OF COMBUSTION OF THE COAL DETERMINED BY THE SOLVAY PROCESS COMPANY AND THE BABCOCK & WILCOX COMPANY

Test No.	B.T.U. per Lb. of Dry Coal*		Ash in Coal by Proximate Analysis		B.T.U. per Lb. of Combustible		B.T.U. per Lb. of Combustible from Ultimate Anal.†	
	Solvay	B.&W.	Solvay	B.&W.	Solvay	B.&W.	Solvay	B.&W.
1	14369	14354	5.26	5.16	15167	15135	15308	15542
2	14225	14224	6.48	6.14	15211	15154	15413	15497
3	14400	14215	5.74	6.21	15277	15156	15677	15322
4	13529	13984	8.01	5.98	14707	14873	15125	14844
5	13747	14045	6.61	5.04	14720	14790	15103	14939
6	13572	14501	6.58	4.60	14528	15200	15159	15648
7	14240	13759	4.85	6.56	14966	14725	15021	15291
8	14009	13920	6.61	5.21	15000	14685	15266	14381
9	13769	14227	8.37	6.12	15027	15154	15233	14988
10	13876	14500	6.85	5.77	14896	15388	15400	15463
11	13889	14232	6.71	5.17	14888	15008	15257	15011
12	13790	14230	7.62	5.30	14927	15026	15472	14682
14	14111	14433	5.84	4.67	14986	15140	15427	14793
16	14155	14797	8.62	5.22	15490	15612	16007	15240
17	14265	14721	7.95	5.81	15497	15629	16244	15910
18	13385	13993	11.01	7.95	15041	15202	15663	15378
Gen. Avg.	13960	14259	7.07	5.68	15020	15117	15423	15183

* Secured with an oxygen bomb calorimeter.

† From formula B.t.u. $= 14600\,C + 62000\,(H - \frac{O}{8}) + 4000\,S$

TABLE 9 RONEY STOKER TEST DATA

Test No.	Date of Trial	Duration of Trial, Hr.	Steam Pressure by Gage, Lb. per Sq. In.			Barometric Pressure, In. Mercury at Temp. of Boiler Room	Average Draft, In. of Water			Temperature, Deg. Fahr.							
			In Boiler	Entering Superheater	Leaving Superheater		Below the Two Dampers	In Furnace	In Ash Pit	Fire Room	Saturated Steam Corresponding to Outlet, Gage Pressure	Superheated Steam	Superheated Steam at Outlet Y	Superheat	Feed Water Entering Boiler	Gases Leaving Boiler	Gases Entering Rear Bank of Tubes	Gases below Damper
1	5-17-18-11	25	195	193.5	190	29.26	0.42	0.32	0.16	82	383.7	519	515	135.3	184	576	903	602
2	5-18-19-11	24	192	190.2	188	29.17	0.05	0.16	0.06	81	382.8	498	497	115.2	180	480	851	479
3	5-20-21-11	24	198	196.1	192	29.23	0.51	0.39	0.21	80	384.5	515	516	130.5	184	542	992	565
4	5-21-22-11	30	201	199.0	192	29.39	0.55	0.22	0.02	81	384.6	521	520	136.4	181	670	1064	709
5	5-21-25-11	24	199	....	187	29.45	0.16	0.24	0.10	73	382.5	490	491	107.5	183	483	847	501
6	5-25-26-11	24	200	197.8	191	29.37	0.57	0.26	0.16	79	384.2	521	515	136.8	185	662	1085	668
16	6-18-19-11	32	193	191.4	188	29.37	0.17	0.23	0.12	79	382.9	485	482	102.1	177	460	678	467
17	6 - 20 - 11	16.5	207	205.5	194	29.39	0.95	0.34	0.06	79	385.4	518	521	132.6	179	636	927	622
18	6-20-21-11	24	207	205.2	194	29.48	1.11	0.33	0.05	73	385.4	543	540	157.6	178	694	1029	709
2-4†	5-18-22-11	90	199	195.7	191	29.27	0.38	0.28	0.06	81	384.1	515	514	130.9	182	572	....	...
5-6†	5-24-26-11	55	209	195.9	189	29.40	0.38	0.25	0.05	76	383.3	509	503	125.7	183	575	....	...

† Includes periods between tests.

TABLE 10 RONEY STOKER. TEST DATA AND RESULTS

Test No.	Relative Humidity of Atmosphere, Per Cent	Feed Water, Lb. per Hr.	Moist Coal, Lb. per Hr.	Dry Coal, Lb. per Hr.	Dry Coal, Lb. per Hr. per Sq. Ft. of Grate	Ash (Wet), Lb. per Hr.	Ash (Dry), Lb. per Hr.	Per Cent Ash based on Dry Coal	Steam used by Stoker Engines, Lb. per Hr.			Steam Used by Jets, Lb. per Hr.		
									Saturated Steam	Equivalent in Superheated Steam	Per Cent of Total Steam	Saturated Steam	Equivalent in Superheated Steam	Per Cent of Total Steam
1	60.3	74208	7615	7463	16.73	586	446	5.98	123	114	0.15	.....	.....	.....
2	64.8	57036	5714	5609	12.58	483	366	6.52	112	105	0.18	.....	.....	.....
3	66.7	80318	8271	8130	18.23	764	602	7.40	141	131	0.16	400	373	0.47
4	64.5	107056	11794	11582	25.97	982	757	6.54	173	161	0.15	1650	1534	1.43
5	62.4	67099	6727	6606	14.81	607	455	6.89	136	128	0.19	1110	1045	1.56
6	66.5	106196	11551	11295	25.32	903	692	6.13	127	118	0.11	1530	1420	1.34
13	51.6	70148	6845	6665	14.92	736	620	9.30	136	128	0.18	860	813	1.16
17	54.4	135676	14107	13761	30.85	1325	1134	8.24	220	205	0.15	1800	1675	1.24
18	55.2	135770	15319	14987	33.60	1750	1470	9.81	195	179	0.13	1750	1614	1.19
2–4	66.0	84401	8991	8834	19.81	781	602	6.81	133	124	0.15	....	....	....
5–6	63.9	90039	9653	9459	21.21	841	647	6.84	132	123	0.14	....	....	....

TABLE 11 RONEY STOKER. COMPUTED RESULTS

Test No.	Factor of Evaporation	Equivalent Water Evaporated into Dry Steam from and at 212 Deg. Fahr., Lb. per Hr.	Equivalent Evaporation from and at 212 Deg. Fahr. per Lb. of Dry Coal	Equivalent Water per Hr. from and at 212 Deg. Fahr. per Sq. Ft. of Boiler Heating Surface	Horsepower Developed	Per cent of Rated h.p. Developed on Basis of 10 Sq. Ft. of Boiler Heating Surface per h.p.	Combined Efficiency of Boiler and Furnace
1	1.1582	85948	11.52	3.63	2491	105.0	77.84
2	1.1514	65671	11.71	2.78	1903	80.0	79.88
3	1.1559	92840	11.42	3.92	2691	113.8	77.45
4	1.1621	124410	10.74	5.26	3606	152.4	75.78
5	1.1441	76768	11.62	3.24	2225	94.0	81.15
6	1.1581	122984	10.89	5.20	3565	150.7	75.28
16	1.1476	80502	12.08	3.40	2333	98.6	80.98
17	1.1625	157722	11.46	6.67	4572	193.3	76.73
18	1.1766	159747	10.66	6.75	4630	195.7	75.57
2–4	1.1581	97745	11.06	4.13	2833	119.8	76.13
5–6	1.1540	103905	10.98	4.39	3012	127.3	76.23

TABLE 12 TAYLOR STOKER. TEST DATA

Test No.	Date of Trial	Duration of Trial, Hr.	Steam Pressure by Gage, Lb. per Sq. In.			Barometric Pressure, In. Mercury at Temp. of Boiler Room	Average Draft, In. of Water			Temperature Deg. Fahr.							
			In Boiler	Entering Superheater	Leaving Superheater		Air Blast in Tuyeres	Suction below Boiler Dampers	Suction over Fire	Fire Room	Saturated Steam Corresponding to Outlet Gage Pressure	Superheated Steam	Superheated Steam at Outlet Y	Superheat	Feed Water Entering Boiler	Gases Leaving Boiler	Gases Entering Rear Bank of Tubes	Gases below Damper
7	6- 2- 3-11	24	205	202.6	193	29.45	1.30	0.58	0.37	73	384.9	540	535	155.1	187	575	959	539
8	6- 3- 4-11	24	200	197.0	192	29.46	0.76	0.20	0.26	70	384.6	513	505	128.4	181	493	748	466
9	6- 5- 7-11	50	206	202.7	192	29.31	1.73	0.53	0.30	69	384.6	541	538	156.4	184	574	934	549
10*A*	6- 8- 9-11	24	197	195.0	191	29.50	0.65	0.20	0.22	73	384.2	517	513	132.8	188	489	732	465
10	6- 7- 9-11	48	195	193.8	190	29.50	0.67	0.20	0.22	71	383.7	518	515	134.3	187	487	756	460
11*A*	6 - 10 - 11	12	210	206.5	188	29.25	2.54	0.83	0.28	90	382.8	549	546	166.2	186	668	1045	626
11	6- 9-11-11	26.5	210	207.5	190	29.25	2.53	0.84	0.26	83	383.7	549	544	165.3	188	651	1027	611
12*A*	6-12-13-11	24	196	194.7	189	29.04	0.80	0.24	0.22	64	383.2	516	516	132.8	177	514	789	470
12*B*	6-13-14-11	24	198	196.1	190	29.21	0.96	0.27	0.23	70	383.7	523	523	139.3	182	555	844	500
12	6-12-14-11	48	197	195.2	189	29.13	0.88	0.26	0.23	67	383.2	520	519	136.8	179	535	816	484
13	6-14-15-11	11.5	197	195.3	178	29.25	2.54	0.85	0.31	63	378.6	545	538	166.4	178	641	987	584
14	6-15-16-11	24	206	203.0	189	29.30	1.56	0.84	0.33	70	383.2	551	544	167.8	177	647	994	598
15	6-16-17-11	24	199	197.3	188	29.28	1.61	0.57	0.30	69	382.8	528	524	145.2	176	561	976	510
7-9†	6- 2- 7-11	109	203	200.6	192	29.30	...	0.45	....	70	384.6	532	523	147.4	184	545	....	...
10-11†	6- 7-11-11	80.5	200	198.6	190	29.66	....	0.44	....	75	383.8	528	525	144.2	187	542	....	....

† Includes periods between tests.

TABLE 13 TAYLOR STOKER. TEST DATA AND RESULTS

Test No.	Relative Humidity of Atmosphere, Per Cent	Feed Water, Lb. per Hr.	Moist Coal, Lb. per Hr.	Dry Coal, Lb. per Hr.	Dry Coal, Lb. per Hr. per Sq. Ft. of Grate	Ash (Wet), Lb. per Hr.	Ash (Dry), Lb. per Hr.	Per Cent Ash Based on Dry Coal	Steam Used by Stoker Engines and Turbines Driving Fans, Lb. per Hr.		
									Saturated Steam	Equivalent in Superheated Steam	Per Cent of Total Steam
7	71.3	105856	11310	11094	27.39	1002	780	7.03	2997	2766	2.61
8	78.6	76052	7770	7622	18.82	954	483	6.34	1987	1854	2 44
9	75.8	113668	12094	11838	29.23	1257	799	6.75	3528	3254	2.87
10*A*	69.0	66618	6866	6736	16.63	826	663	9.84	1888	1760	2.64
10	71.7	65663	6796	6656	16.43	824	659	9.90	1855	1727	2.63
11*A*	52.0	149627	16212	15904	39.27	2051	1668	10.48	5970	5482	3.66
11	60.9	147435	16000	15696	38.75	1843	1499	9.55	5481	5030	3.41
12*A*	72.3	82060	8507	8339	20.59	827	645	7.73	2222	2071	2.52
12*B*	61.0	88098	9143	8968	22.14	899	757	8.45	2483	2309	2.62
12	66.6	85079	8825	8653	21.37	863	700	8.09	2352	2189	2.57
13	65.0	140671	15239	14946	36.91	1445	1212	8.11	5044	4633	3.29
14	67.2	128105	13730	13469	33.25	1367	1173	8.71	4120	3782	2.95
15	70.6	85762	9362	9158	22.61	972	764	8.34	2565	2378	2.77
7-9	75.0	98073	10421	10209	25.21	1099	737	7.22	2848	2634	2.68
10-11	67.4	93522	10075	9868	24.36	1175	945	9.58	3068	2844	3.04

TABLE 14 TAYLOR STOKER. COMPUTED RESULTS

Test No.	Factor of Evaporation	Equivalent Water Evaporated into Dry Steam from and at 212 Deg. Fahr., Lb. per Hr.	Equivalent Evaporation from and at 212 Deg. Fahr. per Lb. of Dry Coal	Equivalent Water per Hr. from and at 212 Deg. Fahr. per Sq. Ft. of Boiler Heating Surface	Horsepower Developed	Per cent of Rated h.p. Developed on Basis of 10 Sq. Ft. of Boiler Heating Surface per h.p.	Combined Efficiency of Boiler and Furnace
7	1.1658	123407	11.12	5.22	3577	151.2	77 07
8	1.1579	88061	11.55	3.72	2553	107.9	80.28
9	1.1695	132935	11.23	5.62	3853	162.8	77.85
10*A*	1.1529	76804	11.40	3.25	2226	94.4	77.63
10	1.1545	75808	11.39	3.22	2197	92.9	77.90
11*A*	1.1719	175348	11.03	7.41	5083	214.8	76.18
11	1.1697	172456	10.99	7.29	4999	211.3	75.84
12*A*	1.1639	95510	11.45	4.04	2768	117.0	79.21
12*B*	1.1624	102405	11.42	4.33	2968	125.5	79.19
12	1.1640	99032	11.44	4.18	2870	121.3	79.24
13	1.1790	165851	11.09	7.01	4807	203.2	76.56
14	1.1822	151447	11.24	6.40	4390	185.5	76.42
15	1.1713	100453	10.97	4.23	2912	123.1	74.90
7–9	1.1648	114235	11.19	4.83	3312	140.0	77.66
10–11	1.1598	108467	10.99	4.58	3143	132.8	75.66

TABLE 15 RONEY STOKER. HEAT BALANCE, PERCENTAGES OF TOTAL HEAT IN COAL

Flue Gas Analyses and Temperature Taken in Breeching

No. of Test	Absorbed by Boiler	Moisture in Coal	Hydrogen in Coal	Heat to Chimney			Carbon Monoxide	Carbon in Ash	Radiation, etc.
				Heat to Chimney	Moisture in Air	Total			
1	77.84	0.18	4.55	13.56	0.37	13.93	0.23	1.20	2.07
2	79.88	0.16	4.36	9.15	0.26	9.41	0.42	1.20	4.57
3	77.45	0.15	4.48	11.39	0.32	11.71	0.74	1.85	3.62
4	75.78	0.17	4.49	12.79	0.36	13.15	1.95	2.13	2.33
5	81.15	0.16	4.29	9.11	0.20	9.31	1.29	2.29	1.51
6	75.28	0.21	4.74	13.17	0.36	13.53	1.16	1.71	3.37
16	80.98	0.22	4.18	8.94	0.19	9.13	0.27	3.20	2.02
17	76.73	0.22	4.47	12.47	0.28	12.75	0.74	2.05	3.04
18	75.57	0.21	4.64	14.34	0.27	14.61	0.62	2.42	1.93
								Average	2.72

Flue Gas Analyses and Temperatures Taken Below Dampers

1	77.84	0.18	4.59	13.83	0.38	14.21	0.0	1.20	2.00
2	79.88	0.16	4.36	9.03	0.27	9.30	0.27	1.20	4.83
3	77.45	0.15	4.58	12.84	0.37	13.21	0.09	1.85	2.67
4*	.....	....	....	.....	....	.....	....	....	...
5	81.15	0.16	4.32	9.43	0.21	9.64	0.73	2.29	1.71
6	75.28	0.21	4.75	13.77	0.36	14.13	0.58	1.71	3.34
16	80.98	0.22	4.20	8.97	0.19	9.16	0.0	3.20	2.24
17	76.73	0.22	4.45	12.67	0.29	12.96	0.97	2.05	2.62
18†	.....	....	....	.....	....	.....	....	....	....
								Average	2.77

* Pyrometer out of order and taken out to be examined.

† Leakage of air into collecting device of gas analyses apparatus.

TABLE 16 TAYLOR STOKER. HEAT BALANCE, PERCENTAGE OF TOTAL HEAT IN COAL

Flue Gas Analyses and Temperature Taken in Breeching

No. of Test	Absorbed by Boiler	Moisture in Coal	Hydrogen in Coal	Heat to Chimney			Carbon Monoxide	Carbon in Ash	Radiation, etc.
				Heat to Chimney	Moisture in Air	Total			
7	77.07	0.18	4.58	11.54	0.28	11.82	1.61	2.31	2.43
8	80.28	0.17	4.28	10.12	0.24	10.36	0.40	1.80	2.71
9	77.85	0.20	4.31	11.21	0.25	11.46	0.44	2.20	3.54
10	77.90	0.18	4.34	11.35	0.26	11.61	0.27	2.77	2.93
11	75.84	0.18	4.47	11.91	0.35	12.26	0.59	3.58	3.08
12	79.24	0.18	4.46	11.05	0.21	11.26	0.16	2.32	2.38
14	76.42	0.18	4.51	13.15	0.27	13.42	0.31	2.57	2.59
								Average	2.81

Flue Gas Analyses and Temperatures Taken Below Dampers

7	77.07	0.18	4.52	12.76	0.30	13.06	0.0	2.31	2.86
8	80.28	0.17	4.23	10.20	0.25	10.45	0.04	1.80	3.03
9	77.85	0.20	4.28	11.33	0.23	11.63	0.08	2.20	3.76
10	77.90	0.18	4.30	10.64	0.24	10.88	0.0	2.77	3.97
11	75.84	0.18	4.40	11.60	0.34	11.94	0.77	3.58	3.29
12	79.24	0.18	4.37	10.22	0.19	10.41	0.0	2.32	3.48
14	76.42	0.18	4.43	13.19	0.27	13.46	0.25	2.57	2.69
								Average	3.29

DISCUSSION

R. H. Rice. The use of large boiler units is a step in the right direction, and it is gratifying to note that the large boiler which forms the subject of these tests has a high efficiency.

I believe that E. D. Leavitt was the first engineer to build boilers in large units, and his 1000-h.p. boilers of the Belpaire type built for the Calumet & Hecla mine, the Metropolitan Water Works of Boston, and elsewhere, are entirely successful units and have high efficiency.

The paper, since it confines itself to a report of the results obtained by testing these large boilers, does not, of course, bring out the advantages of such boilers in the operation of central power stations. It is manifest that by the use of such boilers the fire-room force will be reduced, and the complication of the plant greatly diminished. The floor space occupied is less and the cost of the boiler house will, therefore, be considerably smaller. The question of maintenance and repairs to brick work, stokers, etc. remains to be settled by experience. It would seem by inspection of Figs. 1 to 3 that improvements will have to be made in the design of the brick settings. Fig. 2, especi-

ally, where the Roney stoker has been applied, would seem to leave much to be desired.

The particular type of boiler adopted gives ample room for complete combustion of the gases before they enter between the cold tubes and this is probably an essential to obtaining the highest efficiencies.

This test calls attention to the absurd method of rating a boiler in terms of horsepower. The horsepower unit used was adopted many years ago when uneconomical engines were in use. It would seem that the proper method of rating a boiler would be in thousand pounds of steam per hour at a given pressure and temperature. In this way determination can be made without much computation of the adaptability of the boiler to the units which the steam is to drive.

It should also be pointed out that the test gives no indication of the comparative merits of the two stokers, except so far as their efficiency is concerned. The other items, such as ease of handling, smokelessness, and cost of maintenance, which are essential to a determination of the relative merits of the stokers, do not come within the scope of the paper. It would be interesting to know whether the preliminary tests were extensive enough to establish the best operating condition for each stoker as regards air pressure, depth of fire bed, etc.

In attaching a single boiler to a unit of 6000 or 7000 kw. capacity, problems must arise which are not met with where a battery of boilers is used to supply such a unit. The blowing off of 75,000 or 100,000 lb. of steam into the atmosphere in case of sudden stoppage of the turbine unit is a serious matter, and in such installations means will have to be taken to protect the boiler by immediate injection of a large amount of feedwater in order to reduce the temperature. Tests with a flow meter have shown that a boiler that forms part of a battery can have its output reduced almost to zero in a small fraction of time by the introduction of cold feedwater, and these experiments indicate clearly in the case of such large units the necessity of an arrangement for accurately graduated feed following closely the demands for steam on the boiler, so that the disturbance by feed of the steam capacity of the boiler can be reduced to a minimum while maintaining constant pressure. In these large units fluctuations in feed supply are likely to lead to large fluctuations in pressure when operated as single units. Furthermore, the stresses on the boiler due to expansion forces will be greatly reduced by the use of such graduated feed. There would seem to be a field for invention in this connection.

It is unfortunate that our methods of controlling the elements in boiler operation are so variable that such boiler tests as the ones reported, involve enormous expense and difficulty and a considerable amount of time. The uncertainty and difficulty of the test largely depend on the irregularities of operation of the furnace; and these irregularities are clearly set forth in Par. 17, where it is shown that an error in judgment of the condition of the fires at the end of the test may involve as large quantities as $2\frac{1}{2}$ to 5 tons of coal. It is these irregularities and difficulties which render necessary the testing of boilers for such extended periods, and if our furnaces and grates could be so improved that uniformity of conditions could be quickly estab-

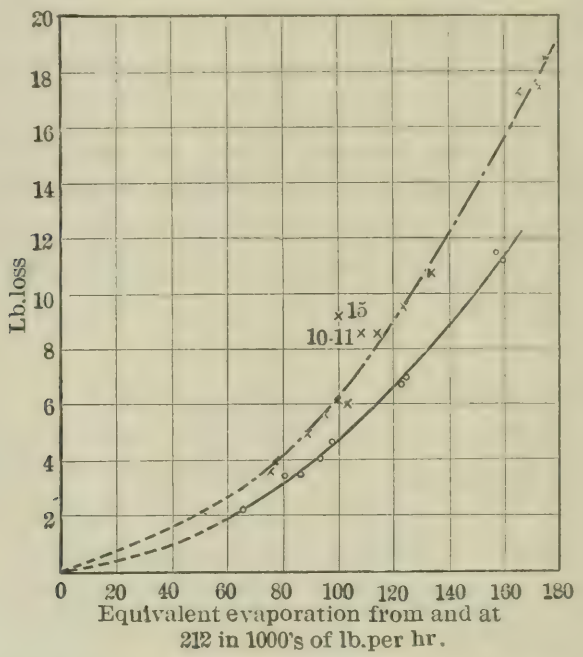


FIG. 12 LOSS OF PRESSURE THROUGH SUPERHEATER AND EQUIVALENT EVAPORATION
Points 15 and 10–11 include banking

lished and maintained, it is manifest that the testing periods could be greatly reduced. Establishment of such uniform conditions would also undoubtedly result in increased efficiency.

R. D. DEWOLF.[1] From the data given in these tests I have prepared some curves which may be of interest. Fig. 12 shows the drop in pressure through the superheater for different ratings of the boiler. The lower scale is not the per cent rating, but gives the equiva-

[1] Asst. Mech. Engr., Rochester Ry. & Light Co., Rochester, N. Y.

lent evaporation from and at 212 deg., in thousands of pounds per hour. Although we have two boilers built by the same company from exactly the same specifications and under as nearly the same conditions as

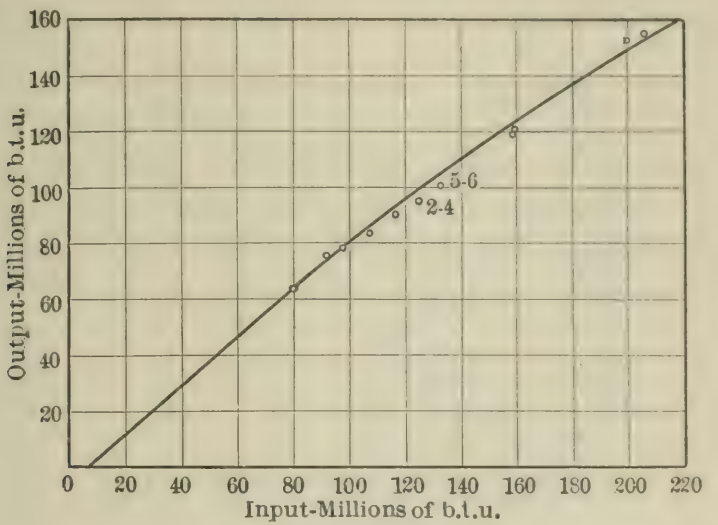


FIG. 13 INPUT AND OUTPUT RONEY STOKER
Points 2–4 and 5–6 include banking

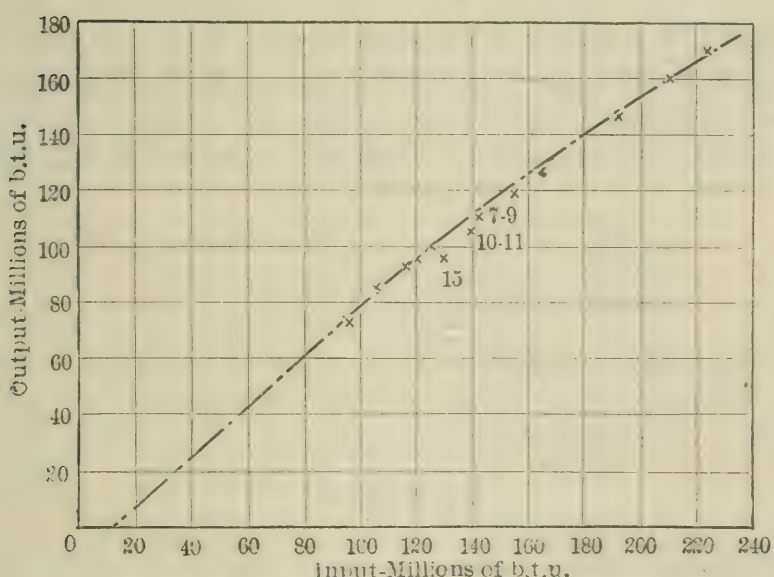


FIG. 14 INPUT AND OUTPUT TAYLOR STOKER
Points 15, 10–11, 7–9 include banking

possible, we get nevertheless a considerable difference in the drop of pressure through the superheater under the same operating conditions. In Fig. 12 the upper curve is taken from the boiler equipped with the

Taylor stoker and the lower curve from the boiler equipped with the Roney stoker.

Fig. 13 illustrates an input-output curve, showing the input in millions of B.t.u. and an output in the same units for the boiler equipped with the Roney stoker. It will be observed that the two

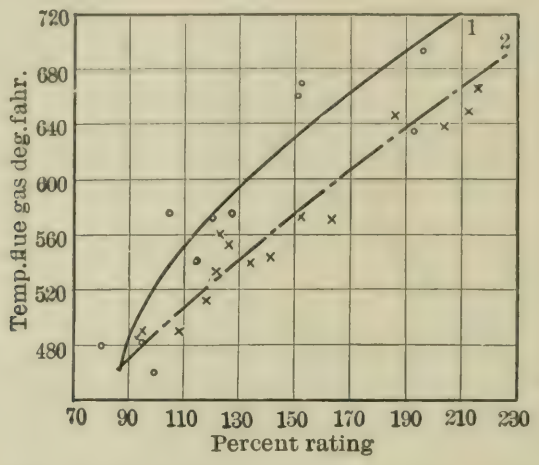


Fig. 15 Per Cent Rating and Flue Gas Temperature
Curve 1 indicates Roney Stoker, Curve 2 Taylor Stoker

points marked 2–4 and 5–6 are somewhat below the line. These points include the periods between tests and are therefore at a somewhat lower efficiency. Fig. 14 shows the same period for the boiler equipped with a Taylor stoker. This also shows that the points in-

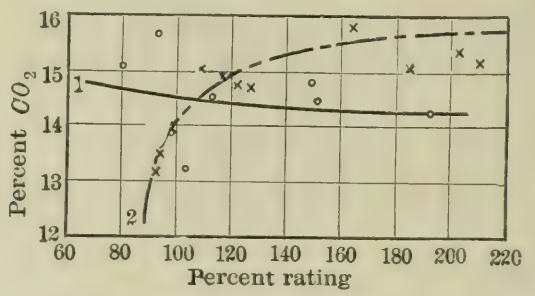


Fig. 16 Per Cent Rating and Per Cent CO_2
Point 15 includes $7\frac{1}{2}$ hours banking. Curve 1 indicates Roney Stoker, Curve 2 Taylor Stoker

cluding the period between tests and the point 15 which includes $7\frac{1}{2}$ hours of banking, are below the curve. Fig. 15 shows the variation in temperature of the flue gas with the rating at which the boiler is operated. The upper curve is taken from the boiler equipped with

a Roney stoker and the lower curve from the boiler equipped with the Taylor stoker. It will be noticed that the points fall more or less generally over the chart. The crosses are the points for the Taylor stoker, and the circles the points for the Roney stoker.

Fig. 16 shows the relation between the per cent of CO_2 and the rating at which the boiler is operated. The dash and dot line is for the boiler equipped with the Taylor stoker, and the full line curve is for the boiler equipped with the Roney stoker. The lower scale gives the percentage of rating at which the boiler is operated. Fig. 17 gives the percentage of steam used by the boiler auxiliary, that is, the stoker engine and the turbine operating the fan for the Taylor stoker. Curve

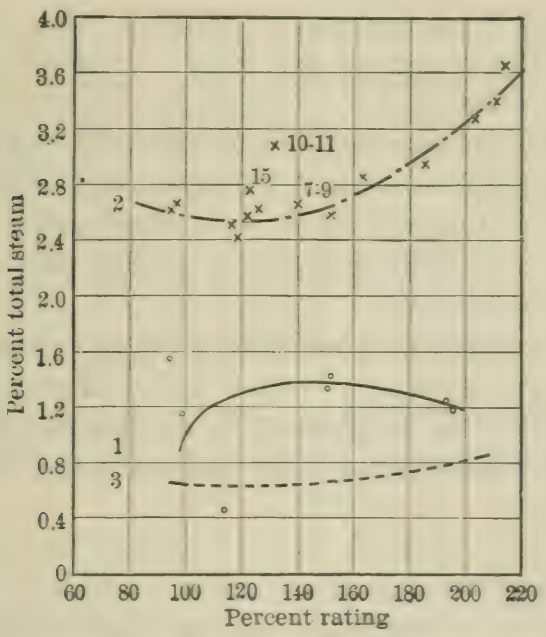


FIG. 17 PER CENT TOTAL STEAM USED AND PER CENT OF RATING
Points 15, 7–9, 10–11 include banking. Curve 1 indicates Roney Steam Jets, Curve 2 Taylor Stoker Engine and Fan Turbine, Curve 3 = 25 per cent of Curve 2

No. 1 represents the steam used by the steam jets in the Roney stoker. The dotted curve is purely a theoretical curve, being taken as 25 per cent of curve No. 2 and is intended to indicate in a general way the loss of heat energy from the steam used by the stoker engine and the turbine for the Taylor stoker equipment. The individual points from which all of these curves are plotted, cover a wide area and the curves themselves are only averaged as nearly as possible.

H. O. Pond. I believe that greater emphasis should be laid in Par. 9 upon the excellent feedwater conditions obtaining at the Delray plant. In a plant where the feed conditions were less favorable the advisability of using such large boiler units would probably be somewhat questionable.

Referring to Par. 17 it seems to me that the substantial agreement in the results on the short tests emphasizes the accuracy with which the tests were made and the accuracy with which balancing at the start and finish were taken care of.

The point which impressed me most in the results obtained in these tests is the small difference in efficiency shown over a range of more than 100 per cent in the rating of the boilers, the difference between the highest and lowest efficiencies obtained being only about 6 per cent. With boilers showing such high sustained efficiencies over this wide range of rating it would seem advisable to change the ordinary method of plant operation, as brought out in Par. 28, making it possible to run all the boilers in the plant all of the time instead of throwing many of them on bank, as is the usual practice now, thus permitting a considerable increase in general station economy and reduction of fixed charges.

In connection with the installation of these boilers and the arrangement for the series of tests recorded in Dr. Jacobus's paper I feel that credit should also be given to F. A. Scheffler of the Babcock & Wilcox Company whose efforts lent largely to the success of the ultimate installation.

Wm. D. Ennis. James Watt, in a curious old letter which I recently ran across, refers to a colossal new boiler which he was then building, which exposed 300 sq. ft. of surface to the fire, whereas his largest previous boiler contained only 170 sq. ft. Dr. Jacobus has managed to get some 24,000 sq. ft. over one fire.

It is not only a big boiler: it is also an efficient one. A 75 per cent running efficiency including a lay-over period of 7 hours, with a load varying from rating to double rating, means a great deal in power station work. With a turbine giving a brake horsepower on 220 B.t.u. per min., and with a brake to switchboard ratio of $1\frac{1}{2}$, it means 26,400 B.t.u. in fuel per kw-hr., about 1.9 lb. of coal.

This economy was obtained, not at a 3 or $3\frac{1}{2}$ lb. evaporation rate but at one running as high as 7 lb. In a city power station the cost of land and buildings and interest on the same will be approximately in the inverse ratio of this evaporation rate. The big boiler is efficient commercially as well as thermally.

What caused this high thermal efficiency? In some respects the operation was not remarkably economical. The loss to the ashpit was not especially low and would have been a conspicuous item had the coal carried a greater percentage of ash. Flue gas temperatures were not particularly low, the ratios of heating surface to grate surfaces (as the latter were measured) being 53 and 58½, certainly not extreme figures. The unaccounted-for radiation loss was low as should be expected with the low ratio of setting surface to heat generated, which must prevail in a large boiler. This has a particularly important influence on the rate of heat transmission, because a com-

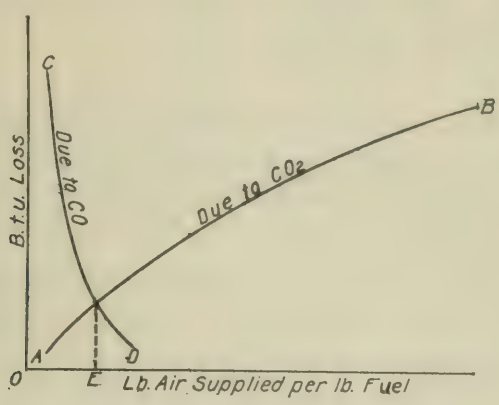


FIG. 18 RELATION BETWEEN AMOUNT OF AIR SUPPLY AND B.T.U. LOSS

paratively slight radiation loss means a considerable reduction in the transmission factor due to temperature gradient through the heating surface.

The explanation of the high furnace efficiency lies in the form and dimensions of the furnace which is exactly adapted to produce a thorough mixture and impingement of the gases. With the very low draft normally carried stratification must have been almost impossible. This is demonstrated by the composition of the gas, which showed a maximum of ½ of 1 per cent of CO, with CO_2 varying from 10.35 to 16.50 per cent. The loss of the fuel heat due to the formation of CO was only 0.16 to 0.38 per cent. When we consider the extraordinary degree of care exercised in the tests, these figures are notable. It is not very easy to find the CO actually present in a rapidly moving stream of gas. Roughly $\frac{1}{10}$ of 1 per cent of CO in the gases means about 65 B.t.u. loss per pound of coal (based on a 14 lb. air supply). This is about as much as the loss due to an increase in flue gas temperature of about 20 deg. It is as much as the loss due to a decrease of 1½ per cent in the CO_2.

In the great majority of plants where a large air excess seems to be the normal and almost inevitable condition of operation, the best percentage of CO_2 is the highest obtainable. This does not hold for such advanced practice as Dr. Jacobus's paper represents. The situation then is somewhat as suggested by the sketch. As the air supply decreases, the loss due to excess air (CO_2 loss) also decreases, but at some certain point which may be different for every furnace and every condition of operation, the formation of CO begins to be an important offsetting factor. The air supply for the best efficiency is that determined by the minimum point of the combined curves *AB* and *CD* (Fig. 18). This point has been pretty closely approximated in the tests reported in the paper.

H. de B. Parsons. In order to secure perfect combustion, three requirements must be observed, namely: a surplus of air, a thorough mixture of the fuel particles with the oxygen in the air, and a high temperature. A furnace that fails to offer any or all of these conditions will not support perfect combustion and without perfect combustion, the efficiency of the boiler and furnace cannot be high.

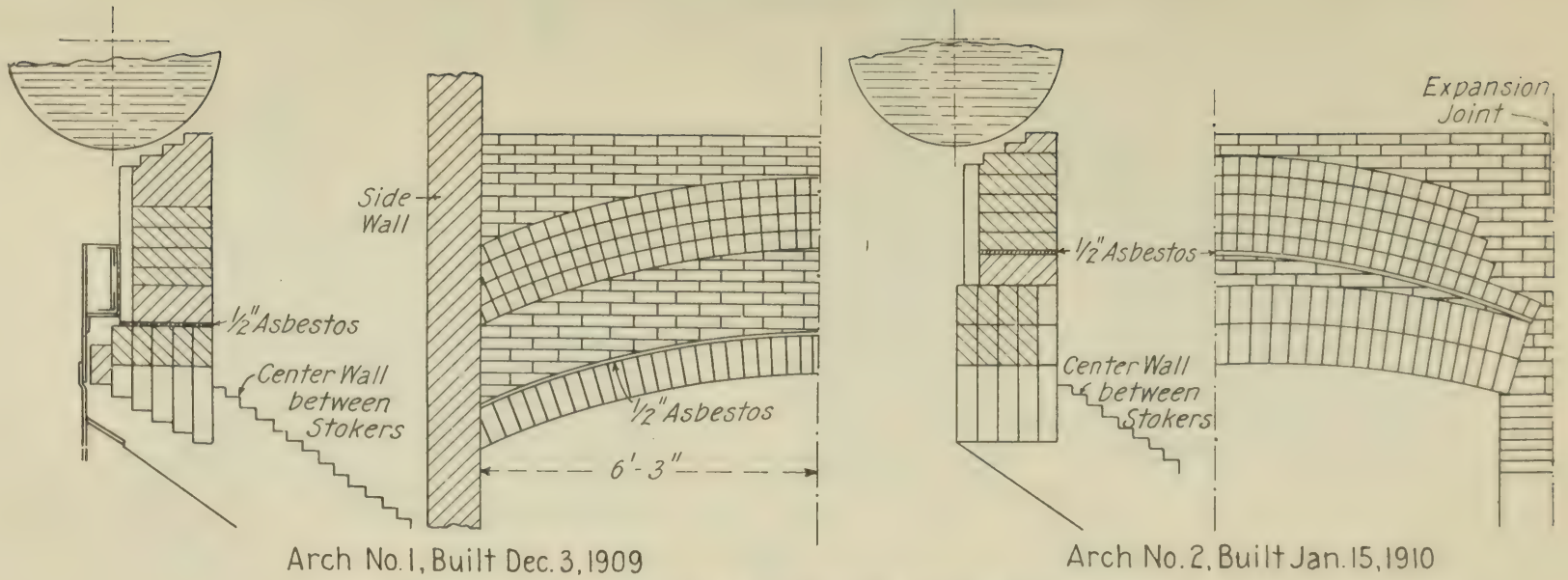
In the intense heat of a furnace, the bituminous portion of the coal is vaporized with such great rapidity, that it is practically impossible to burn all the gaseous portion before it is drawn into the chimney by the draft, or before it is cooled by contact with the water surfaces and has its temperature reduced below that for ignition.

The object of the furnace is to generate heat, while the object of the boiler is to rob the furnace of its heat. The speaker wrote, over 15 years ago, "in some designs the products of combustion are not admitted to the heating surfaces until after combustion has become complete. This arrangement has met with success in many instances, and could be carried much farther than it is."

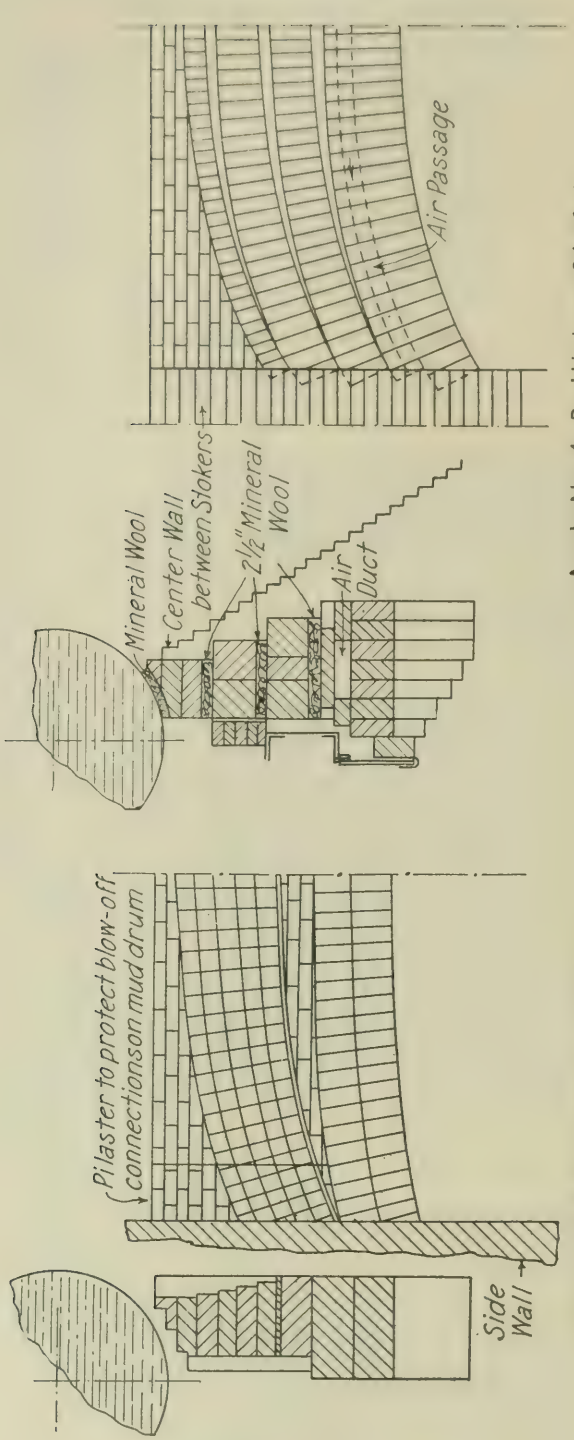
With the boilers described by the author, this is the practical result obtained. The fire is so far from the heating surfaces and the combustion chamber is so large, that a thorough mixture of oxygen and fuel can be maintained and combustion completed before the products are made to pass over the water surfaces. It is not difficult to see why this result is so, when we notice that the distance from the grate to the tubes is about three times that in ordinary boilers.

A study of the flue gas analyses illustrates the success obtained in mixing the oxygen with the fuel, and the degree of perfection of combustion as shown by the small percentage of carbon monoxide present.

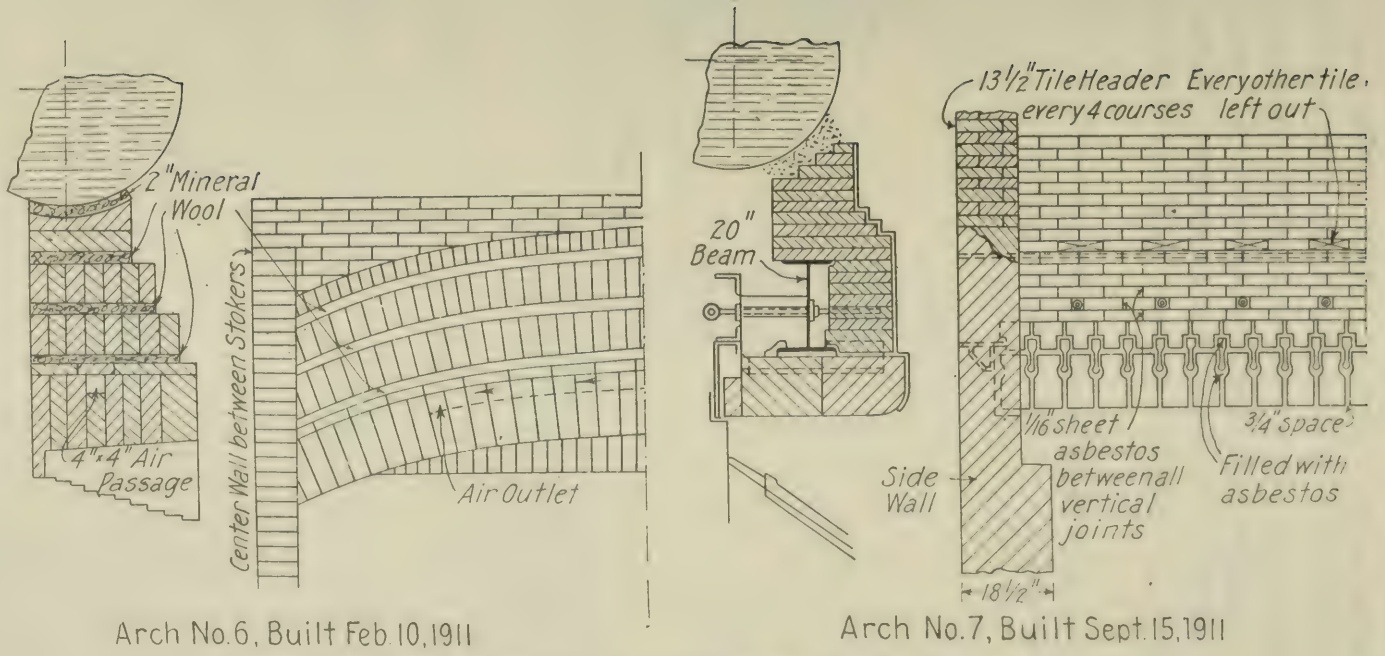
Figuring the air supply from the gas analyses, the result shows that



FIGS. 19A AND B TYPES OF ARCHES USED OVER RONEY STOKERS


Pilaster to protect blow-off connections on mud drum
Side Wall
Arch No. 3, Built March 1, 1910
Mineral Wool
Center Wall between Stokers
2½" Mineral Wool
Air Duct
Air Passage
Arch No. 4, Built June 24, 1910

FIGS. 19 C AND D TYPES OF ARCHES USED OVER RONEY STOKERS


2" Mineral Wool
4" x 4" Air Passage
Center Wall between Stokers
Air Outlet
Arch No. 6, Built Feb. 10, 1911
20" Beam
13½" Tile Header Every other tile every 4 courses left out
1/16" sheet asbestos between all vertical joints
3¼" space
Filled with asbestos
Side Wall
18½"
Arch No. 7, Built Sept. 15, 1911

FIGS. 19E AND F TYPES OF ARCHES USED OVER RONEY STOKERS

the surplus air was only about 27 per cent of the theoretical amount. The quantity of air supplied per pound of coal was between 14 and 14.5 lb. These are very low figures and could only be the result of using large grates and by the most careful management of the fires.

There must always be a loss of heat in the gases passing to the flue. Under the conditions described in the paper the theoretical maximum efficiency attainable would have been about 87 per cent. Therefore the efficiency ratio, that is the ratio between the actual and the theoretical efficiency, was about 89 per cent, certainly a remarkable showing.

The author and the designers of the boilers are to be congratulated on the performance.

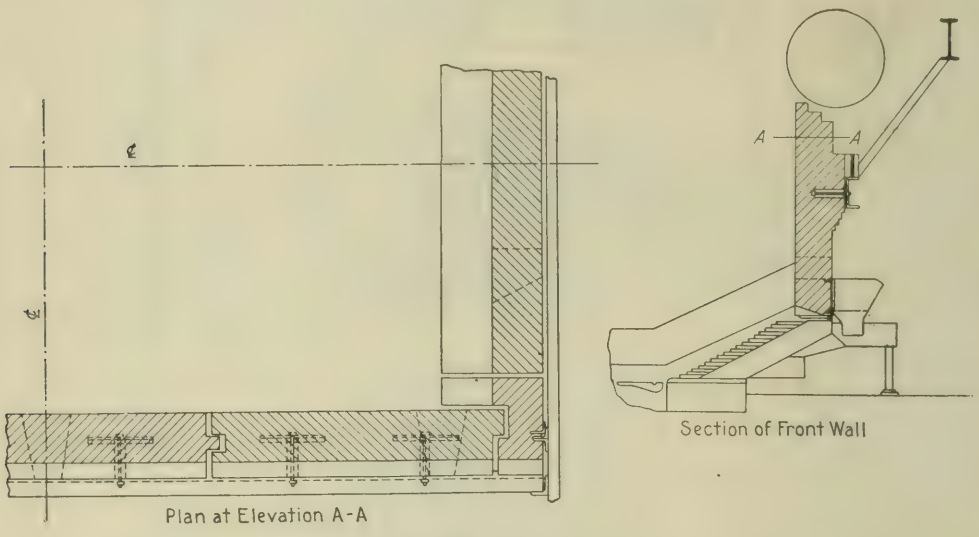


FIG. 20 DETAIL OF NEW SETTING FOR TAYLOR STOKERS

HORACE H. ESSELSTYN. In connection with these large boilers, as was expected we had trouble with the brick work. Fig. 19 shows some views of the arches which were tried out before the flat arch was finally adopted. Most of these views have to do with the Roney stoker. Fig. 20, however, shows how the bulging of the front wall in the frame work of the setting for the Taylor stoker was overcome by anchoring the front wall of the frame work on the boiler. This setting has now been in service for five months and during that period has caused no trouble.

Fig. 19a-f shows the different arches that were tried on the Roney setting, and the date gives the approximate life of each arch. It must not be understood that the arch failed at exactly that time but it was evident in each case that it was advisable to change the construction. Arch No. 7 is the last and is now working very successfully.

E. G. Bailey. The tests are very interesting, especially with respect to the high boiler efficiency and low loss due to radiation, etc. These two items of the heat balance, however, are those which show the greatest variations should there be any error in the sampling of the coal burned or the determination of its heating value.

The precaution of having different samples analyzed in the laboratories of two large and reputable concerns, taken by the author, would seem ample in this connection, but a careful study and comparison of the individual and average analyses bring out some points worthy of mention.

Of the 26 sets of check analyses given in Tables 6 and 7, there are only two cases where the ash reported by the Solvay laboratory is lower than that reported by the B. & W. From the average results of the two laboratories as given in Table 8, we note the Solvay reports 1.39 per cent higher ash and 300 less B.t.u. than the B. & W. On individual samples the difference between the results of the two laboratories is as great as 3.40 per cent ash and 929 B.t.u., or over 6 per cent of the heating value. The ash from different samples of the same kind of coal tested in the same laboratory varies 6.16 per cent and the heating value 1015 B.t.u., or 7.3 per cent. While each lot of coal shipped under any one trade name or even from a single mine does not run absolutely uniform, yet such variations as these are so exceptional that they cast some doubt upon the accuracy of the sampling and analysis. Even the B.t.u. per pound of combustible, which is practically a constant for coal from a given mining district, varies 750 B.t.u., or 5 per cent, as determined by one of the laboratories, while the other reports results in which this factor varies 600 B.t.u., or 4 per cent. The variation in this factor as determined by the two laboratories from the same original sample shows variations of similar magnitude.

The erratic variations in the ash indicate that sufficient care was not used in taking and working down the samples, as the accurate determination of ash from a pulverized sample is a very simple matter. Properly taken duplicate samples should have checked in each individual instance within 0.50 per cent ash, and the averages should have been within 0.10 per cent. There are other differences in the volatile, heating value, and ultimate analyses that indicate greater variations in the laboratory work which are brought out by the comparison in Table 17. The averages from the two laboratories are shown and distinction is made between the two coals. The Red

Jacket coal covers tests 1 to 12, 14 and 18, while Milburn coal was used on tests 16 and 17.

Comparison is also made in Table 18 between the results of the two laboratories on Red Jacket coal and also some results of the United States Geological Survey and West Virginia Geological Survey on coal from these and adjoining mines on a combustible basis, for the purpose of showing variations in the laboratory analyses and B.t.u. determinations.

As it is perfectly evident that there is no satisfactory agreement between these two sets of samples or analyses, the question arises as to which, if either, is right. A check is available on the ash by taking the weight of dry ash as weighed up from the boiler tests and correcting it for the percentage of combustible. The two laboratories agreed with reasonable closeness on the moisture and combustible determinations from the ash:

Laboratory	Combustible	Moisture
Solvay	29.85	20.42
B. & W.	26.29	20.93

The average net dry ash as determined by this method is 5.53 per cent from Red Jacket coal and 6.30 per cent from the Milburn. While there is always some ash carried away with the escaping gases, it would probably be a small factor with this particular setting. If the average of 6.90 per cent ash as determined by the Solvay laboratory were correct, the ash carried away by the gases would be 20 per cent of the total ash, while the B. & W. average of 5.71 per cent corresponds to a loss of 3 per cent from the grates. If any appreciable amount of ash were carried away by the draft it should be more pronounced on the tests with the higher rates of combustion, but the plotting of these results fails to indicate any such relation. On the other hand, the weighed up net ash on the six tests of Red Jacket coal where the rate of combustion was below 20 lb. per sq. ft. of grate per hour is 5.38 per cent, while on the three tests having a rate of combustion above 30 lb. it is 6.62 per cent of ash. The general conclusion to be drawn from these data is that the true average ash in the coal used was about $\frac{1}{2}$ per cent lower than the average of the two sets of laboratory determinations.

The fact that the B.t.u. determinations were made in an oxygen bomb calorimeter does not in itself insure accuracy in the results. Table 18 shows the heating value per pound of combustible, corrected for sulphur, as determined in the two laboratories, to be about 3

per cent below the corresponding results as made by the United States Geological Survey and the West Virginia Geological Survey. The latter results agree closely with the work of other laboratories that have had their calorimeters and thermometers calibrated by the United States Bureau of Standards. The disagreement in the ulti-

TABLE 17 COMPARISON OF AVERAGE ANALYSES OF CHECK SAMPLES BY TWO LABORATORIES

ALL DATA TAKEN FROM TABLES 6, 7 AND 8

	RED JACKET COAL		MILBURN COAL	
Laboratory	Solvay	B. & W.	Solvay	B. & W.
Moisture	1.83	2.07	2.53	2.55
Dry basis				
Volatile	34.04	31.86	28.28	27.77
Fixed carbon	59.06	62.43	63.43	66.71
Ash	6.90	5.71	8.29	5.52
B.t.u	13922	14187	14210	14759
Carbon	79.11	77.14	80.47	80.06
Hydrogen	5.03	5.86	5.34	5.68
Oxygen	6.41	8.97	3.92	6.82
Nitrogen	1.13	1.11	1.26	1.07
Sulphur	1.09	1.09	0.62	0.86
Ash	7.23	5.83	8.39	5.51
B.t.u. per lb. combustible by bomb calorimeter	14953	15045	15493	15620
B.t.u. per lb. combustible calculated	15323	15127	16125	15575

TABLE 18 COMPARISON OF ANALYSES OF COAL FROM SAME SEAM BY DIFFERENT AUTHORITIES, COMBUSTIBLE BASIS

Laboratory	Solvay	B. & W.	U. S. Geological Survey	W. Va. Geological Survey
Volatile	36.56	33.76	35.61	36.28
Fixed carbon	63.44	66.24	64.39	63.72
B.t.u.[1]	15082	15128	15500	15599
Carbon	86.29	82.87	86.48	85.07
Hydrogen	5.49	6.30	5.17	5.14
Oxygen	6.99	9.64	6.77	8.53
Nitrogen	1.23	1.19	1.58	1.26

mate analyses as made in the different laboratories indicates that very little confidence is to be placed in the heating value calculated from this analysis.

[1] Corrected to sulphur free basis.

As further evidence of errors in the heating value determinations as reported in the paper, the B.t.u. from Tables 17 and 18 have been plotted in Fig. 21 with radiation, etc., from the first set of heat balances of Tables 15 and 16. While the percentage of radiation is probably affected to some extent by other factors, it is very evident from the location of these points that the real heating value of the coal used on the various tests was much more uniform than the figures used in the calculations. The lower the B.t.u. used in the calculations, the higher would be the efficiency and the lower the radiation loss, and vice versa.

The errors in the sampling and analysis of the coal are evidently much greater than those entering into the other results of the tests. If Dr. Jacobus had had all the coal used on each test check-weighed

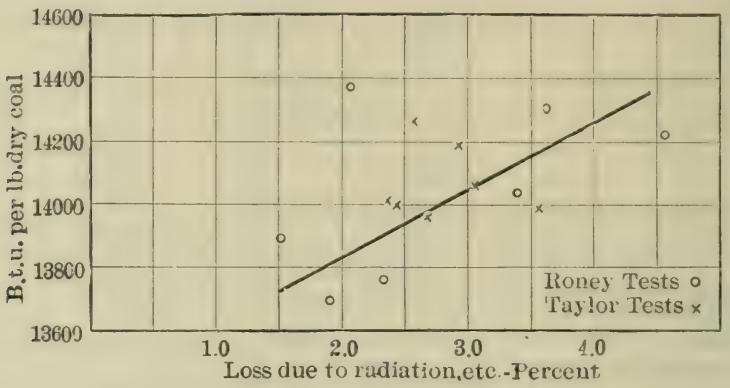


Fig. 21 Relation of Radiation Loss to B.t.u. of Coal

on two sets of scales by two observers, and they had not agreed more closely than 6 per cent, would he have averaged the weights with only a passing comment and no attempt at an explanation? There would undoubtedly have been some rigid investigation and the cause of the difference soon determined and rectified. If the efficiency of the boiler and furnace are to be determined, it is just as important to know the true heating value of the coal as it is to know the weight of coal and water used during the test.

By averaging the results of the two laboratories, Dr. Jacobus has evidently assumed that they each made errors of equal magnitude in opposite directions. It would be very strange if such were the case when the differences are so great. There is strong evidence to the contrary, and it appears that the average heating value used in calculating the efficiencies and heat balances is about 3.5 per cent low, thereby causing a corresponding error in the efficiency and radiation.

The author has accredited the high efficiency and low radiation to the type of boiler and furnace, but it seems that the low heating value used in the calculations is a very important factor which should receive due consideration before these results are accepted as final proof of such marked superiority of this type of boiler equipment.

William Kent. The paper shows a most important advance in the art of construction and setting of boilers, and is the record of a remarkable series of long tests giving the highest efficiencies at high as well as low rates of driving ever obtained from coal containing over 25 per cent of volatile matter.

The most notable thing about the setting is the enormous size of the combustion chamber. About 12 years ago I wrote concerning the existing and earlier practice in boiler setting as follows: "Thirty or

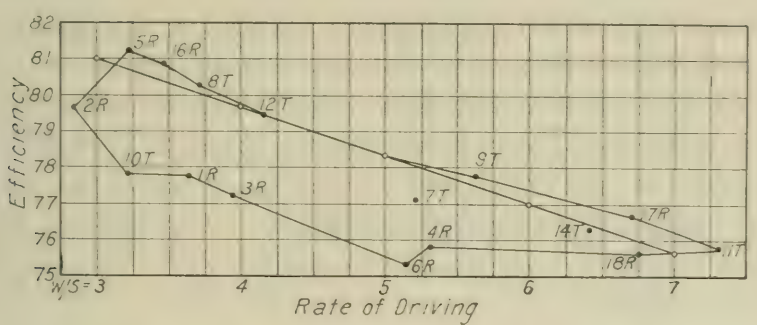


Fig. 22 Relation of Efficiency to Rate of Driving

forty years ago it used to be the custom to set boilers with the grate surface near to the shell of the boiler, 12 to 15 in. being a common distance. . . . A distance of 2 to 3 ft. from the grate to the boiler is now common practice for bituminous coal. With very smoky coal 4 ft. is sometimes used; and 6 to 8 ft. would be better." These boilers in Detroit have the grates about 10 ft. below the lowest point of the heating surface and the total height of the furnace, as stated by Dr. Jacobus, is about 29 ft. That this is good proportioning, worthy of being followed in the future, is evidenced by the results obtained in the tests.

Concerning the advisability of large boiler units, we can only admire the courage of the Detroit Edison Company, in making such a great advance over former practice, and watch the results with interest. It would be interesting a few years hence to get a statistical record of the numbers of times each boiler was thrown out of service for brickwork and tube troubles and for cleaning, and the length of

time that elapsed after each stoppage before the boiler was running again at full pressure. The size of the unit emphasizes the necessity of the best possible brickwork, tubes and workmanship of the best quality, treatment of the feedwater before it enters the boiler, and the utmost precautions to prevent the entrance of oil or grease.

As to the economy or efficiency obtained in the tests, I have made a somewhat extensive study of the report, and have reached some conclusions that seem to be of importance. In the first place, I have plotted the relation of the efficiencies to the rate of driving (Fig. 22) using the 16 long and complete tests, 9 with Roney and 7 with Taylor stokers. The line drawn all around the plotted points shows the field covered by them. Many years ago I plotted in like manner the field of the results of the boiler tests made at the Centennial Exhibition in 1876, and made the remark that the upper boundary of the field represented the highest results that could be expected, when every condition was most favorable, and the breadth of the field (vertically, it was very broad) represented the depth of our ignorance as to what were the best conditions and how they might be obtained. At the time those tests were made there was no Orsat or Hempel apparatus for analyzing the flue gases, and no one knew just what composition of gas was coincident with high efficiency. The field of the Detroit tests is much narrower, and we have the gas and the coal analyses to aid us in our study.

The inclined straight line on the diagram corresponds to the equation

$$E = 81 - 1.33\left(\frac{W}{S} - 3\right)$$

in which E = the efficiency in per cent, W = lb. of water evaporated from and at 212 deg. per hr., and S = sq. ft. of heating surface. This line represents approximately the relation of the efficiency to the rate of driving for the 10 best results of the 16. Comparing this line with the 16 tests we find 6 Taylor tests near the line, 1 (No. 10) about 2.8 per cent below it; 4 Roney tests near the line, 5 from 1.4 to 2.8 per cent below.

How much of the deviation of the six tests that are considerably below the line is due to errors of the test or to errors in the sampling, analysis, or calorimetric determination of the heating value of the coal? The errors of the test are probably insignificant; each test was at least 24 hours long, and it is scarcely possible that the difference in quantity and quality of coal in the furnace at the beginning and end

of the test amounted to as much as a quarter of an hour's coal supply, which would make an error of 1 per cent in the efficiency.

The analyses and calorimetric values show errors of considerable magnitude, which may be due either to inaccurate sampling or to inaccurate apparatus and methods. The following show some of the largest discrepancies:

Moisture in Coal	No. 2	No. 4	Ash in Coal	No. 7	No. 12A
Solvay	1.30	1.94	Solvay	4.85	9.03
B. & W.	2.36	1.66	B. & W.	6.56	5.58

B.t.u. per unit Combustible

		By Calorimeter	By Analysis	Difference
No. 8	Solvay	15,000	15,266	+266
	B. & W.	14,685	14,381	−304
	Difference	−315	−885	
No. 17	Solvay	15,497	16,244	+747
	B. & W.	15,629	15,910	+281
		+132	−334	
No. 12	Solvay	14,927	15,472	+545
	B. & W.	15,026	14,682	−344
	Difference	+99	−790	
Average of 16 Tests	Solvay	15,020	15,423	+403
	B. & W.	15,117	15,183	+ 66
		+97	−240	

The heating value by calorimeter, average of Solvay and B. & W. figures, was used in the computations of efficiency. The greatest difference is in No. 6, Solvay, 14,528; B. & W., 15,200; difference 672. The average figure may possibly be 336 from the true value, or 2.2 per cent, but it is more probable that the Solvay figure is too low and the B. & W. figure too high, so that the average of the two figures is likely to be within 1 per cent of the true value. The maximum probable errors of the test, of the analysis for moisture and ash, and of the heating value may balance each other for any single test, or they may be added together making a total error of as much as $2\frac{1}{2}$ per cent or nearly as much as the maximum deviation of any test from the straight line on the diagram. If there were no other causes

for the deviation we might say that the deviation from the line was due to the errors named, but there is another most important one, namely, the difference in the air supply as shown by the analysis of the flue gases, which we will now consider, assuming that the possible errors above referred to do not exist, that the efficiencies as recorded are correct, and that the analyses of the gases in the flue are accurate. These analyses compared with those of the gases taken from the bottom of the last pass show that there was some leakage of air into the setting, but it was not serious. The conclusions in Table 19 from the computations resulting from the analyses of the gases drawn from the flue would not be materially changed if they had been made from the analyses of the gases drawn from the bottom of the last pass.

TABLE 19 RELATION OF DIFFERENT VARIABLES TO EFFICIENCY

No. of Test	$\frac{W}{S}$	E	H	K	t	f	a
1*R*	3.63	77.84	4.55	15,150	302	20.98	146
2*R*	2.78	79.88	4.36	15,180	302	17.59	257
3*R*	3.92	77.45	4.48	15,220	305	19.12	228
4*R*	5.26	75.78	4.49	14,790	304	16.65	261
5*R*	3.24	81.15	4.29	14,760	310	17.25	165
6*R*	5.20	75.28	4.74	14,860	305	16.99	260
16*R*	3.40	80.98	4.18	15,550	304	18.60	165
17*R*	6.67	76.73	4.47	15,560	306	17.09	204
18*R*	6.75	75.57	4.64	15,120	312	17.73	182
7*T*	5.22	77.07	4.58	14,850	312	17.60	192
8*T*	3.72	80.28	4.38	14,840	315	18.39	138
9*T*	5.62	77.85	4.31	15,090	316	17.13	208
10*T*	3.22	77.90	4.34	15,140	313	21.12	152
11*T*	7.29	75.84	4.47	14,950	301	16.31	212
12*T*	4.18	79.24	4.46	14,980	316	18.33	155
14*T*	6.40	76.42	4.51	15,060	313	17.79	173
Average				15,069	309		194

One peculiar thing is noticeable in these analyses. In 13 out of 16 tests the percentage of CO was higher in the flue than it was at the bottom of the last pass and at the top of the last pass, but in only one case was it higher at the top than at the bottom of the last pass. In test No. 7, the CO was 0.08 at the bottom of the last pass, 0 at the top and 0.42 in the flue. This indicates errors in the analysis, which are not surprising, as it is difficult in the ordinary use of either the Orsat or Hempel apparatus to get accurate figures for CO.

In studying the relation of the air supply to the efficiency, Table 19 was used, which contains the principal figures taken from the report

and some results computed from the figures. The letters in the table have the following meaning: R, Roney stoker, T, Taylor stoker, $\frac{W}{S}$ rate of driving, or lb. of water evaporated from and at 212 deg. per hr. per sq. ft. of heating surface, E, efficiency per cent, H, loss of efficiency due to hydrogen in coal, K, heating value of the coal per unit combustible, t, temperature of saturated steam minus temperature of the boiler room, f, lb. of dry gas per lb. of carbon burned, calculated from the formula $f = \frac{11\mathrm{CO_2} + 8\mathrm{O} + 7\,(\mathrm{CO} + \mathrm{N})}{3\,(\mathrm{CO_2} + \mathrm{CO})}$, a, coefficient of performance described below.

The values of a are found from the formula[1]

$$a = \frac{K - tcf}{1 + R\frac{S}{W}} - \frac{K(E+H) \times (K - tcf)\,S}{970\,c^2 f^2\,W}$$

In this formula c is the specific heat of the gases, taken as 0.24, and R the radiation in units of evaporation (1 unit of evaporation = 970 B.t.u.) per sq. ft. of heating surface per hour, taken in this case at 0.05, which corresponds to a radiation loss equal to $1\frac{1}{2}$ per cent of the heat absorbed by the boiler when it is driven at a rate of 3 units of evaporation per sq. ft. of heating surface per hour.

The coefficient, a, or the criterion of performance, means that when the efficiency is obtained as a result of a test, all the variables that that can be measured, such as K, t, f, S, W, and H being known, and the values of R and C estimated as nearly as possible, then a is a coefficient of performance covering all the other variables whose values are not measured, such as imperfect combustion, resistance of the heating surface and its coatings of soot and scale or grease to transmission of heat, short-circuiting of the gases, velocity of the gases, etc. In this case it covers also the loss of efficiency due to carbon in ash. The value of a in tests that show high efficiency is usually between 200 and 400; the higher the efficiency, other things (such as air supply) being equal, the lower the value of a. The theory upon which the formula is based, is given at length in Steam Boiler Economy,[2] but it may be considered without reference to theory as an empirical formula to be used as a criterion of boiler performance.

In these tests the values of a run from 138 to 261, averaging 194. Let us trace the relation that may exist between the different values of a and other variables.

[1] Steam Boiler Economy, William Kent, p. 220.

Kind of stoker. All of the seven tests with the Taylor stoker give what may be called low values of a, from 138 to 212, averaging 176. The Roney stoker gave values ranging from 146 to 261, averaging 208. The high values are coincident with deficient air supply and are not due to the stoker itself.

Rate of driving. Twelve out of the 16 tests give values of a from 138 to 212, the rate of driving $\frac{W}{S}$ in these tests covering the whole

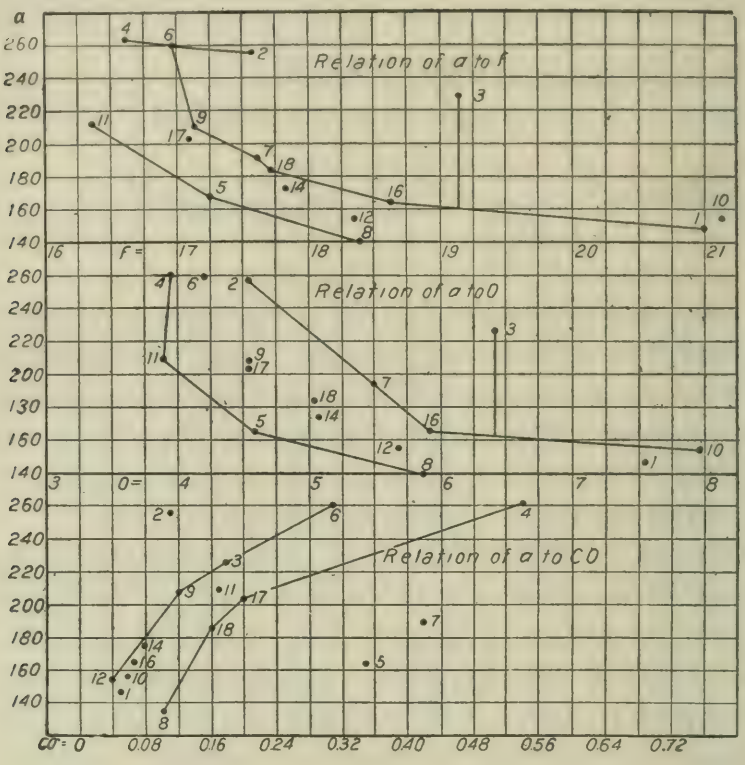


FIG. 23 RELATION OF a TO f, O AND CO

range from 3.22 to 7.29. The four tests giving higher values, 228 to 261, cover the range of $\frac{W}{S}$ from 2.78 to 5.26. The value of a thus appears to be independent of the rate of driving, and this indicates that the velocity of the gases along the heating surface, within the limits of these tests, is not one of the factors of efficiency. In test No. 11, with $\frac{W}{S} = 7.29$ and efficiency = 75.84 per cent, the velocity of the gases is more than double that in No. 10 with $\frac{W}{S} = 3.22$ and efficiency = 77.90 per cent.

Percentage of oxygen in the dry gases, and weight of dry gases per pound of carbon burned. With the exception of one erratic test, No. 3, a approaches its minimum values when the oxygen is above 5 per cent, and when the weight of gas per lb. of carbon burned is above 17.5 lb. (see Fig. 23). The reason for this is apparently that below these figures there is more or less imperfect combustion, as is evidenced by the increasing per cent of CO in the gases.

Per cent of CO in the gases. In Fig. 23, the relation of the value of a to CO is plotted. Excepting Nos. 2, 5 and 7, the plotting shows a somewhat definite relation between these two variables. Values of a below 212 are all found with CO = 0.20 or less, except No. 7, which gives $a = 192$ for CO = 0.42, and No. 5 which gives $a = 165$ for CO = 0.35, but this may be due to an error in the analysis or in the record. In the case of No. 7 the per cent of CO in the bottom of the last pass

TABLE 20 THEORETICAL EFFICIENCIES OBTAINED FROM CONDITIONS OF DETROIT TESTS

	$\frac{W}{S}$ =	3	4	5	6	7
$a = 200, f=$	16	82.20	81.17	80.03	78.81	77.53
$f=$	18	80.08	78.68	77.15	75.53	73.87
$f=$	20	77.81	76.00	74.02	71.94	69.87
$a = 150, f=$	18	81.40	80.44	79.35	78.17	76.96
$a = 250, f=$	18	78.75	76.92	74.95	72.88	70.79

is 0.08 and in the top of the last pass 0. All the tests with the Taylor stoker with this exception gave less than 0.20 for CO in the flue gases, and all gave a less than 212. In the Roney tests the CO covers the whole range from 0.05 to 0.52, and the value of a the whole range from 146 to 261.

Other variables. According to the formula from which the values of a are derived the efficiency depends chiefly on the rate of driving $\frac{W}{S}$, and on the air supply or the pounds of dry gas per pound of carbon, f. It also depends, but to a much smaller extent, on K, t and R. The differences in the values of these variables in these tests are not great enough to make any material difference in the values of a and they do not explain the erratic values found for Nos. 2, 3, 5 and 7. These are possibly due to errors in the analyses for ash and moisture, which would make the reported efficiencies inaccurate and lead to wrong values of a.

The writer's formula for the relation of efficiency to the rate of driving, to the air-supply and to the other measurable variables, upon which boiler efficiency depends, is

$$E + H = \frac{K - tcf}{K\left(1 + \frac{RS}{W}\right)} - \frac{970\,ac^2f^2W}{K(K - tcf)S}$$

Taking as the conditions which approximately correspond to those of the Detroit tests, $H = 4.5$, $K = 15{,}000$, $t = 300$, $c = 0.24$, $a = 150{,}200$ and 250, $f = 18$, and $\frac{W}{S} = 3$, 4, 5, 6 and 7, the theoretical efficiencies in Table 20 are obtained.

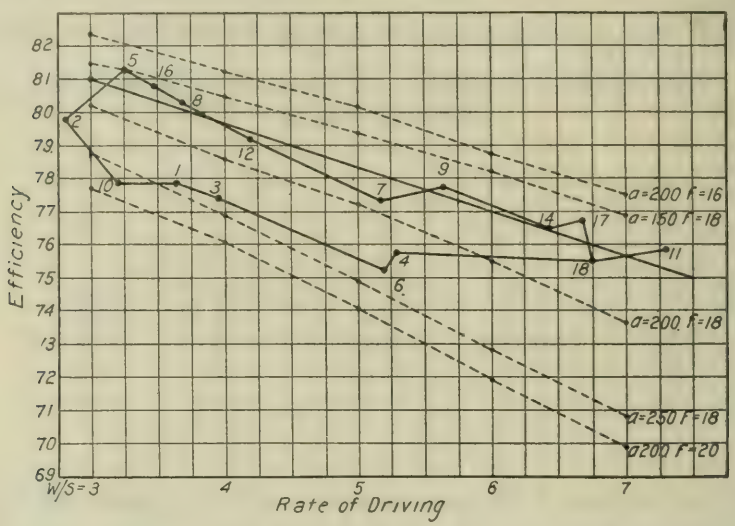


FIG. 24 EFFICIENCY CORRESPONDING TO DIFFERENT VALUES OF f AND a

These figures are plotted in five curves in Fig. 24, and the efficiencies obtained in the Detroit tests are plotted inside of the field covered by the curves. For $\frac{W}{S}$ above 3 the curves are approximately straight lines; somewhere below 3 they begin to curve downwards, so as to reach 0 when $\frac{W}{S} = 0$. The several straight lines to which the curves approximate may be represented by the following formulae:

$$a = 200,\ f = 16,\ E = 82 - 1.17\left(\frac{W}{S} - 3\right)$$

$$f = 18,\ E = 80 - 1.55\left(\frac{W}{S} - 3\right)$$

$$f = 20,\ E = 78 - 2.00\ \left(\frac{W}{S} - 3\right)$$

$$a = 150,\ f = 18,\ E = 81.5 - 1.11\left(\frac{W}{S} - 3\right)$$

$$a = 250,\ f = 18,\ E = 79 - 2.00\ \left(\frac{W}{S} - 3\right)$$

The straight line corresponding to the ten best Detroit tests, the formula being $E = 81 - 1.33\ (\frac{W}{S} - 3)$, is also shown in Fig. 24.

The fact that the ten best of the Detroit tests all lie within the narrow field bounded by the curves marked $f = 18$, $a = 200$ and $f = 18$, $a = 150$, whose maximum width where $\frac{W}{S} = 3$ corresponds to only 1.32 per cent range of efficiency, and where $\frac{W}{S} = 7$ to 3.09 per cent, is a highly satisfactory confirmation of the usefulness of the writer's formula and of the curves drawn from it. The formula itself is rather cumbrous, but when the radiation loss is small, as it is in very large boilers when driven above their normal rating, the simpler straight line formulae and curves drawn from them may be used instead of it. The straight line formulae are valid only when $\frac{W}{S} > 3$.

The most important conclusion to be drawn from the set of curves in Fig. 24, is the great importance of keeping down the air supply at high rates of driving, so that f does not exceed 18. When $\frac{W}{S} = 7$, an increase of f from 18 to 20 corresponds to a decrease of efficiency from 73.9 to 69.9 per cent. The drop in efficiency increases much more rapidly if f increases beyond 20, and if the rate of driving is increased beyond $\frac{W}{S} = 7$. An increase of f from 18 to 20, causes a greater decrease of efficiency than an increase of a from 200 to 250. The need of gas analysis to control the regulation of the air supply becomes of great importance at high rates of driving.

The curves in Fig. 24 may be used in answering the question whether it pays to drive boilers in excess of 3 lb. evaporation per sq. ft. of heating surface per hour, the approximate point of maximum fuel economy, since they show what loss in efficiency may be expected at the higher rates of driving with different assumed values of f and a.

Where coal is cheap and when boilers are to be run at maximum capacity for only a few hours a day, the loss in fuel by rapid driving may be more than offset by the reduction in interest on the investment when one boiler is made to do the work usually expected of two or

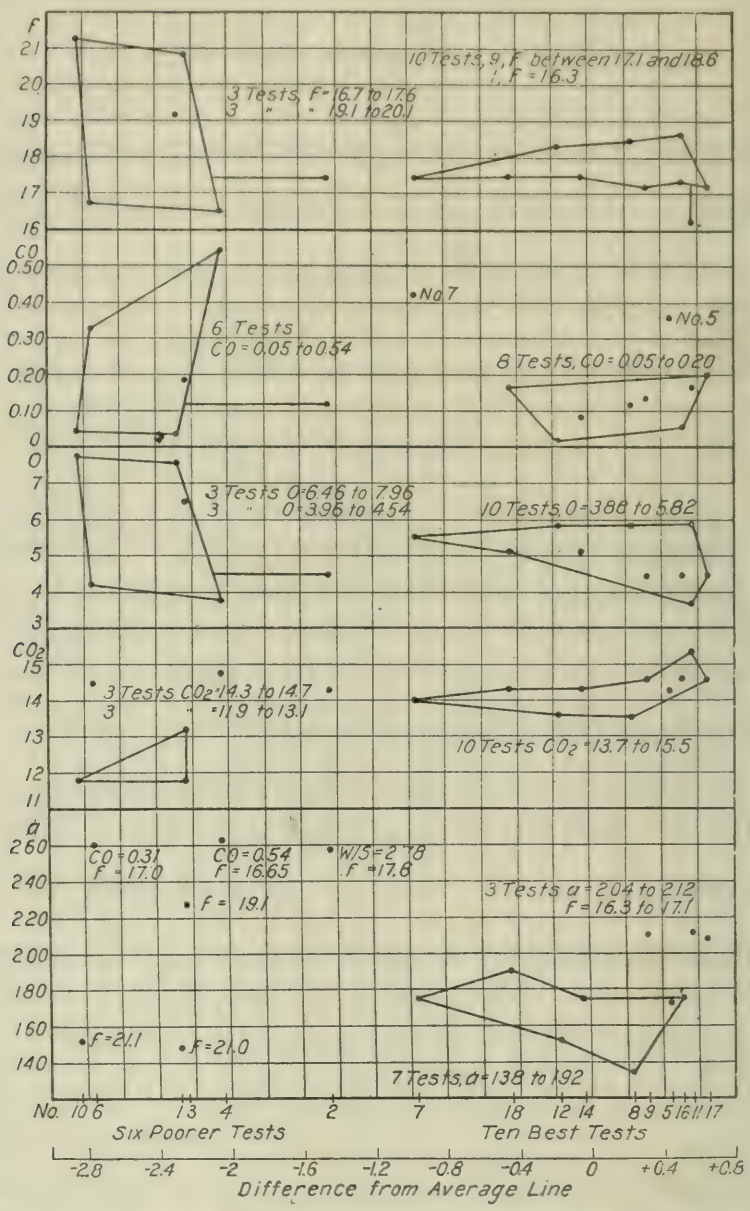


FIG. 25 RELATION OF THE VARIATION OF EFFICIENCY TO f, CO, O, CO_2 AND a

more. The Detroit tests give us definite figures of the reduction in efficiency due to increased rates of driving when a good grade of bituminous coal, very low in moisture, is used. With semi-bituminous coal the lines showing decrease of efficiency would be less inclined

from the horizontal, but with poorer grades of bituminous coals, high in volatile matter and moisture[1] the line would be steeper, indicating still greater decrease of efficiency at high rates of driving.

Taking the formula $E = 81 - 1.33\,(\frac{W}{S} - 3)$ as representing the average of the ten best Detroit tests and comparing the efficiencies actually obtained at the several rates of driving with those given by the formula, and arranging them in the order of the rate of driving, the results in Tables 21 to 24 are obtained.

TABLE 21 TEN BEST RESULTS

No. of test	5*R*	16*R*	8*T*	12*T*	7*T*	9*T*	14*T*	17*R*	18*R*	11*T*
$\frac{W}{S}$	3.24	3.40	3.72	4.18	5.22	5.62	6.40	6.67	6.75	7.29
E by formula	80.68	80.47	80.04	79.43	78.04	77.56	76.47	76.11	76.00	75.28
E by test	81.15	80.98	80.28	79.24	77.07	77.85	76.42	76.73	75.57	75.84
Difference	+0.47	+0.51	+0.24	−0.19	−0.97	+0.29	−0.05	+0.62	−0.43	+0.56

TABLE 22 SIX LOWER RESULTS

No. of test	2*R*	10*T*	1*R*	3*R*	6*R*	4*R*
$\frac{W}{S}$	2.78	3.22	3.63	3.92	5.20	5.26
E by formula	81.29	80.71	80.16	79.77	78.07	77.99
E by test	79.88	77.90	77.84	77.45	75.28	75.78
Difference	−1.41	−2.81	−2.32	−2.32	−2.79	−2.21

Plotting the relation of the differences, that is the deviation of the efficiencies obtained in the several tests from an average line drawn through the plotting of the ten best tests, we reach the following conclusions (see Fig. 25):

Value of f. The ten best results were obtained with f between 17.1 and 18.6 except test 11 *T*, in which f was only 16.31. In this test $\frac{W}{S} = 7.29$, the highest rate of driving. The O in the flue gases was only 3.86, yet the CO was only 0.17, a most exceptional analysis, showing nearly perfect combustion with a very small excess of air-supply over the theoretical requirement of 11.52 lb. air per lb. of carbon. Of the six low results, three were obtained with low values of f, from 16.7

[1] Steam Boiler Economy, p. 225.

to 17.6, three with high values, 19.1 to 21.1. In order to obtain the highest efficiency, therefore, it seems necessary to keep f at or below 18.6; but if it is below 17.6 the efficiency may be either high or low, depending on the percentage of CO in the gases.

Value of CO. Eight of the ten best results were obtained with CO from 0.04 to 0.20; of the two others, No. 5 gave 0.35, and No. 7, 0.42

TABLE 23 TEN BEST RESULTS

No.	Difference	$\frac{W}{S}$	f	a	O	CO	CO_2
17*R*	+0.62	6.67	17.09	204	4.55	0.20	14.69
11*T*	+0.56	7.29	16.31	212	3.86	0.17	15.45
16*R*	+0.51	3.40	18.60	165	5.92	0.07	14.66
5*R*	+0.47	3.24	17.25	165	4.54	0.35	14.40
9*T*	+0.29	5.62	17.13	208	4.57	0.12	14.74
8*T*	+0.24	3.72	18.39	138	5.82	0.10	13.69
14*T*	−0.05	6.40	17.79	173	5.08	0.08	14.20
12*T*	−0.19	4.18	18.33	155	5.73	0.04	13.79
18*R*	−0.43	6.75	17.73	182	5.04	0.16	14.16
7*T*	−0.97	5.22	17.60	192	5.50	0.42	14.00
Average			17.62	179	5.06	0.17	14.38

TABLE 24 SIX LOWER RESULTS

No.	Difference	$\frac{W}{S}$	f	a	O	CO	CO_2
1*R*	−2.32	3.63	20.98	146	7.55	0.05	11.95
3*R*	−2.32	3.92	19.12	228	6.46	0.18	13.05
10*T*	−2.81	3.22	21.13	152	7.56	0.06	11.86
Average			20.41	175	7.32	0.10	12.29
2*R*	−1.41	2.78	17.59	257	4.54	0.11	14.33
4*R*	−2.21	5.26	16.65	261	3.96	0.54	14.74
6*R*	−2.79	5.20	16.99	260	4.23	0.31	14.66
Average			17.08	259	4.24	0.32	14.57

CO. These may be due to errors in the analysis, as in both cases the CO in the flue was much higher than it was either at the bottom or top of the last pass. This might possibly have been due to the burning of carbon particles in the gas to CO between the bottom of the last pass and the flue, but it is not probable. Of the six low tests two were

due to high CO, 0.31 and 0.54 and consequent imperfect combustion. In the other four, CO was low, 0.05 to 0.18 but in three of them the low efficiency was due to excessive air supply, f being 19.1 to 21.1; in the fourth, No. 2, CO, O and f are all low and the cause of the high value of a, and of the efficiency being 1.41 below the line is not apparent. It is probably due to errors of analysis.

Value of O. The ten high results were obtained with O, from 3.86 to 5.82. Of the six low results, three were obtained with high O, 6.46 to 7.96, and three with low O, 3.96 to 4.54. To obtain the highest efficiency, therefore, the O in the flue gases must be below 6, but if it is below 4.5 the efficiency may be low, on account of imperfect combustion.

Value of CO_2. The ten high tests gave CO_2, 13.7 to 15.5, with O, 3.96 to 4.54; three of the six low tests gave CO_2, 14.3 to 14.7, the other three low tests gave low CO_2, 11.9 to 13.1, corresponding to high O, 6.46 to 7.96. It therefore appears that high efficiencies require CO_2

TABLE 25 CALCULATED EFFICIENCIES BY FOUR FORMULAE

W / S	3	4	5	6	8	10	12	14
Detroit	81	78.67	78.34	77.01	74.35	71.69	69.03	66.37
Wyoming	80	78.58	77.16	75.74	72.90	70.06	67.22	64.38
Cincinnati	79	77.62	76.24	74.86	72.10	69.34	66.58	63.82
Centennial	79	77.0	75.0	73.0	69.0	65.0	61.0	57.0

to be not lower than 13.7 but that the efficiency may be low with CO_2 as high as 14.7 if O is low and CO high. A CO_2 indicator is therefore a very useful instrument for showing when the CO_2 gets below 13.7, when low efficiency is obtained, but if it shows 14 to 15 there is no assurance that efficiency is not low on account of low O and high CO. An oxygen indicator would be much more desirable, for if O is between 4.5 and 6, CO will be low and f not above 18.6, which are the conditions for the highest efficiency.

Value of a. The ten high results give a, 138 to 212, the three above 192 giving low O, 3.86 to 4.57, and low f, 16.31 to 17.13. Of the six low results, three gave a high, 257 to 261, with low O, 3.96 to 4.54, low f, 16.65 to 17.59, and in two cases high CO, 0.31, 0.54, evidence of imperfect combustion. The third case is $2R$, with low CO = 0.11, and $a = 257$, an exceptional result, as before noted. The other three low results gave a low to medium, 146 to 228, with high O and f, and low CO.

A further study of causes of deviation of the results of single tests from the average line of the ten best tests might be made by considering the differences in carbon in the ash and in unaccounted for loss, but as these are not great such study would not modify the conclusions already drawn.

For the purpose of comparing the Detroit tests with the results of other series of tests that have shown high records, the writer has taken

TABLE 26 ACTUAL RESULTS COMPARED WITH FORMULAE

Wyoming	No. 1	No. 2	No. 3	No. 4	No. 6	No. 5	
$\frac{W}{S}$	3.88	6.43	9.03	10.52	10.52	14.76	
Efficiency by formula	78.74	75.12	71.44	69.68	69.68	63.29	
Efficiency actual	74.39	73.95	72.06	72.80	69.18	63.25	
Difference	−4.35	−1.17	+0.62	+3.12	−0.50	−0.04	
Cincinnati	No. 1	No. 2	No. 3	No. 4	No. 6	No. 5	No. 7
$\frac{W}{S}$	5.18	5.57	8.42	8.75	9.58	10.07	13.67
Efficiency by formula	75.99	75.45	71.42	71.07	69.92	69.24	64.28
Efficiency actual	74.8	77.9	70.4	70.6	68.2	70.1	64.5
Difference	−1.19	+2.45	−1.12	−0.47	−1.72	+0.86	+0.22
Centennial	Root	Smith	B. & W.	Smith	Galloway		
$\frac{W}{S}$	2.586	2.785	2.791	3.739	4.178		
Efficiency by formula	79.83	79.43	79.42	77.52	76.66		
Efficiency actual	79.30	78.07	77.51	78.20	75.95		
Difference	−0.53	−1.36	−1.91	+0.68	−0.71		

the tests of Babcock & Wilcox marine boilers for the United States Cruiser Cincinnati, and the United States battleship Wyoming,[1] which were made with semi-bituminous coal. He has also selected the five best tests with anthracite egg coal made at the Centennial Exhibition in 1876. From the plottings of these tests the following formulae

[1] See Annual Report, Ch. of Bureau Engrg., 1900; Trans. Am. Soc. M. E., vol. 32, 1139; Steam Boiler Economy, Wm. Kent, p. 394; Industrial Engineering, March, 1911, p. 175.

have been derived; the straight line formula of the Detroit tests is also given for comparison:

Detroit. . . $E = 81 - 1.33\left(\frac{W}{S} - 3\right)$ Cincinnati. . . . $E = 79 - 1.38\left(\frac{W}{S} - 3\right)$

Wyoming. $E = 80 - 1.42\left(\frac{W}{S} - 3\right)$ Centennial. . . $E = 79 - 2.00\left(\frac{W}{S} - 3\right)$

Using these formulae for calculating efficiencies at different rates of driving, the results as given in Table 25 are obtained.

The results calculated by means of the formulae compare as shown in Table 26 with the results obtained in the tests (the comparison for the Detroit tests having already been given).

The Centennial tests still remain the best standard of comparison for anthracite coal. The five tests selected above are the best out of the economy and capacity tests of thirteen different boilers. The results of these five tests have never been excelled with anthracite in any tests reported by competent authority.

The Cincinnati and Wyoming tests had one advantage over the Detroit tests: they were made with semi-bituminous coal. They also had one disadvantage: the restricted space in the furnace for combustion of the volatile gases. Had the Cincinnati and Wyoming tests been made with the bituminous coal used in the Detroit tests they would probably have given considerably lower figures, and had the Detroit tests been made with semi-bituminous coal the figures might have been about 1 per cent higher, on account of the coal being lower in hydrogen and oxygen.

The Detroit tests are remarkable for the uniformity with which the oxygen in the chimney gases was kept so low, in all but three or four tests, as to keep the chimney loss at very low figures on account of the difficulty of obtaining complete combustion of coal high in volatile matter in furnaces with small combustion chambers; and at the same time have almost perfect combustion. Dr. Jacobus might inform us what means were taken to insure the air-supply being regulated with such uniformity.

R. C. Carpenter. I have made many tests of boilers of the same construction and provided with stokers essentially similar to one of these tested, and I have never obtained results so remarkably good. I might also state that I never tested such large boilers.

I have concluded that the high results obtained were due to the method of operating the furnace and to the very high grade of coal

used, together with the remarkable control of the air supply. The flue gas analyses show that practically all of these tests were run with the minimum air supply thus producing maximum CO_2. The air supply was such that there was a little CO in the discharge flue. These conditions, I believe, tend to maximum economy.

In a series of experiments which happened to relate to the burning of cement in a kiln, where the condition of air supply was under control and could be varied rather more than in a boiler which I once made, I found that the critical condition for highest economy was when a small amount of CO appeared in the discharge gases, approximately as shown in the test of the Detroit boilers.

It is interesting to note that the same remarkable control of the air supply was maintained, whether the fires were banked or were being pushed to the extreme limit.

There are two or three points in the tests regarding which further information would be acceptable. I note that the particular setting of this boiler is such that discharge gases can pass through the heating surface on either side. I would naturally expect under that condition that there would be a great difference of draft on the two sides. The maximum air current would sometimes bring a draft on one side, sometimes on the other. I should also expect that the temperature of discharge on the different sides of the boiler might be different. I think it would be desirable for Dr. Jacobus to give us information as to what he found concerning the regularity of draft and temperature on these two sides, and also whether he recorded the difference in temperatures found on the two sides.

It would be interesting also if he would tell us the temperature of the gases in the different passes through the boilers. He gives the temperature of the gases in the discharge, but not at various passes.

E. J. Billings. In view of the value and extremely interesting features of the paper which has been presented, it is hoped that a few criticisms of the analytical work will not be out of place.

In Table 8, it is noted that the average of the heat units per pound of combustible as found by one laboratory agrees with the corresponding calculated values very closely. The work of the other laboratory does not agree so well. Taking the column headed "B. & W." under "B.t.u. per pound of combustible" and omitting tests No. 16 and No. 17 (which were made on a different kind of coal), we find the average to be 15,045 B.t.u. and the mean deviation from the average to be 170 B.t.u. In a similar manner the average of the

"Solvay" column is found to be 14,953 B.t.u. and the mean deviation from the average to be 150 B.t.u.

While these mean deviations are both larger than the conditions seem to warrant, the conclusion is nevertheless that the cause for the wide disparity in the individual analyses, particularly the ash and B.t.u. determinations of the two laboratories, must be sought in the sampling and preparation of the final samples rather than in the actual analytical work.

Particular mention might be made of test No. 6 in which the B.t.u. per pound of dry coal as determined by the two laboratories, shows a variation of 929 B.t.u. or a variation of 464 B.t.u. from the average of the two results. Had the results of either laboratory been used in calculating the heat balance, the boiler efficiency would have been altered through addition or subtraction by 2.5 per cent, thereby introducing in all probability the largest error of the test.

It has been demonstrated in previous proceedings of this Society that close checking in laboratory work is with reasonable care not only possible but to be expected even under more adverse conditions of sampling than those prevailing on these tests.

For the benefit of those who are particularly interested, it is hoped that the author will be able to throw some additional light on this part of his paper.

J. W. Thomas. I have been disappointed in not being able to put before the Society the results of tests on boilers very similar to the Detroit boilers which are being installed at the plant of the Dayton Power and Lighting Company at Dayton, O. These boilers are of the same general construction as the Detroit boilers and have the same characteristics, but are only half the length and have a capacity of approximately 1200 b.h.p.

With regard to the building required to house boilers, in the case of the boilers referred to at Dayton the room was restricted, as they had to go into an old boiler house, but we were able to get 3600 rated h.p. in a floor space approximately 45 by 64 ft. In these boilers the grate area is about 20 per cent larger in proportion to the size of the boiler than in the case of the Detroit plant. They are equipped with Taylor stokers having six retorts to a side, making twelve retorts to each boiler, with an extended grate of 18 in. on either side, which makes our fire box a little less than 18 ft. across and 13 ft. long. In the operation of these boilers we are getting with approximately 1 in. of draft about 175 per cent rating.

Another feature that is noticeable is that the fires on both sides of the furnace have a tendency to pull apart, that is, the fire does not meet in the center of the combustion chamber, but pulls back and clings very closely to the sides of the boiler tubes. This fact may explain the great capacities which we get out of the boiler and possibly another explanation is that the fire on the first pass which contains nearly 60 per cent of the heating surface clings very closely to the tubes.

A. BEMENT. These tests are especially valuable in presenting new and authoritative data on some of the features which influence and determine efficiency. The matters in which I am specially interested are the radiation, unaccounted for losses, and efficiency.

TABLE 27 AVERAGE HEAT BALANCE

	Boiler with	
	Roney Stoker 9 Tests	Taylor Stoker 7 Tests
Heat absorbed by boiler	77.83	77.80
Heat lost by moisture in coal	0.19	0.18
Heat lost by hydrogen in coal	4.47	4.42
Heat lost to chimney (dry gas)	11.67	11.48
Heat lost by moisture in air	0.29	0.26
Heat lost in carbon monoxide	0.82	0.54
Heat lost by carbon in ash	2.01	2.51
Heat lost by radiation	2.72	2.81
	100.00	100.00

The radiation loss being determined by difference, as a remainder, has resulted in loss due to incomplete combustion other than that of carbon monoxide, being included with that of radiation. Thus radiation has been made to appear larger than it is. Sometimes, however, when the engineer has felt doubtful or troubled about the magnitude of the figure, he has reported it as "radiation and unaccounted for." But the unaccounted for portion of the item has not always been given sufficiently serious thought. Owing to the great difficulty, or in fact the practical impossibility of obtaining radiation directly, it has been conceded that under the circumstances the best or only way was by difference, which in itself should be a simple matter provided all other losses are determined. Unfortunately, the incomplete combustion loss other than that of carbon monoxide, has thus far defied

any known analytical method. Thus it has not been possible to make a separation of the losses, or to get even an approximate idea of the magnitude of either. If we could enjoy a condition of complete combustion in a sufficient number of cases, so that a reasonable radiation value could be devised, it would be of great assistance. But such a favorable condition seldom or never exists, especially with bituminous coal.

It is therefore particularly gratifying to find in the large number of valuable tests described in the paper, that the incomplete combustion loss is at a minimum, as demonstrated by the fact of very high efficiency, impossible of attainment, if this loss had not been small.

An average of the author's heat balances is given in Table 27, from which radiation appears as 2.72 and 2.81, or an average of 2.76 per cent, which is quite different from the 6 to 12 per cent supposed by many to represent radiation.

In some very carefully conducted tests on water-tube boilers[1] in which an ultimate analysis of the ash and refuse was employed and in which the loss in dry chimney gas was calculated upon a more exact basis than required by the Society's code, radiation appeared as 2.26 per cent. In these tests the boiler ran at about 150 per cent of rating, the same as the Delray boilers at Detroit. All of the unaccounted for loss was considered as radiation, it being assumed that combustion was complete. But, notwithstanding the fact that the stoker was a chain grate under a tile roof, I believe that combustion was not entirely complete. So with the Detroit boilers I believe that there was incomplete combustion, other than carbon monoxide, as much possibly as 1.5 per cent. If this is true radiation would appear a little over 1 per cent. In this connection, Mr. R. S. Hale analyzed a large number of tests some years ago, with the result that he put radiation at 1 per cent, and as an average an imcomplete combustion loss other than carbon monoxide of 7 per cent. Tests with anthracite coal, hand-fired directly under the exposed tubes of a horizontal water-tube boiler, have shown radiation and unaccounted for loss as low as 3 per cent. If Mr. Hale's estimate is correct, the Delray figure for radiation should be about one-half or three-quarters of 1 per cent, on account of the relatively smaller setting exposure.

It would be of great assistance if we had a carefully devised factor or factors to use in representing radiation. These might be derived

[1] Journal West. Soc. Engrs., vol. 13, p. 209, and Trans. Am Inst. Chem. Engrs., vol. 2.

by difference, from experiments using stoker fired anthracite, burned in an ample or effective furnace space. We would then have a means by which some measurement of incomplete combustion loss other than carbon monoxide could be made, as it could then be obtained by difference.

It is not my intention to infer that incomplete combustion loss is high in the performance with the Delray boilers, but just the opposite, as these performances are most excellent. I consider incomplete combustion to be exceptionally low as compared with general performance, the reason for which is probably to be found in the long travel of gases from grate to heating surface, and the mixing action of the two gas bodies from the opposite stokers as they come in contact with each other.

When very high efficiencies are reported by competent and reliable engineers, two questions are liable to arise. One concerns the accuracy of the heating power determination of the coal; the other the amount of moisture that leaves with the steam. In the past, efficiency in a great many tests has appeared high, due to the calorimetric method giving a result too low, without the engineer being aware of the fact. With these Detroit tests, however, the heating power determination appears to be without question. From such random calculations as I have made, without knowing the source of the coal, the pure B.t.u. values appear to be consistent, and I see no cause for doubt. It would add value to the paper if the author would state what coal calorimeter was employed, also from what state and seam the coal which was used in each of the tests came.

The fact that large amounts of water may leave a boiler with the steam, without being detected[1] by any calorimetric method thus far used, has often contributed to an apparently high efficiency. In the case of these Delray boilers the steam was superheated, so any question of this character is eliminated. Thus it is evident that a very high efficiency was secured. It is not necessary to call attention to the fact that when such large quantities of coal and water are handled as those in these tests that measurements and determinations are very accurate.

The average efficiency of 16 tests as reported is 77.82 per cent. This, however, is on the basis of coal as fed, not coal burned. That we may get an adequate idea of what this efficiency means it should be put upon the basis of heat absorption, because one of the objects

[1] Trans. Am. Soc. M. E., vol. 26, p. 312.

of the tests was to determine the various heat absorbing capacities of the boilers. There was an average incomplete combustion loss to carbon monoxide and carbon in the ash of 2.94 per cent. If this had been burned, as it should have been and if the true possibilities of the heat absorbing capacity of the boilers had been determined, efficiency would have been 80.11 or 2.94 per cent higher. This, however, neglects any incomplete combustion loss other than that of carbon monoxide. If this incomplete combustion loss was 1.5 per cent, as I have ventured the opinion, the efficiency would be 1.5 per cent higher still, or 81.31. Therefore 81.31 instead of 77.82 would be the measure of efficiency of these boilers at the mentioned rate of operation. This does not seem to be out of harmony, viewed in the light of results with oil fuel with which the attainment of complete combustion is less complicated than with coal.

A feeling that prime movers have been increasing in size has prevailed for several years, accompanied by a desire for similarly large boilers. As a matter of fact, the prime mover instead of increasing in size has been decreasing. Before the days of direct connection in electric stations, the engines, belts, shafts, and dynamos occupied much larger floor space than the boilers. Now, a turbo-generator and its auxiliaries occupy less floor area than do the boilers which supply it, notwithstanding the fact that boilers are driven twice as hard as previously. The real change that has taken place may be termed one of intensity, of which, as far as the turbine is concerned, the most important factor is speed. This intensity of operation, while not acting to enlarge size, has been most instrumental in increasing power development. Thus the unit of power compared with the unit of size is very much larger today than formerly. This enlargement of power has come about through a more intense action and any important increase of power per unit of space with boilers must be largely of the same nature, as the following will show. Let us assume a horizontal water-tube boiler of 5000 sq. ft., requiring a setting dimension size of 20 ft. by 14 ft. Let it be set two in a battery and five batteries. These five batteries will occupy 2800 sq. ft. of floor space. There will be four aisles between the batteries, which if assumed to be 7 ft. wide, will occupy 560 sq. ft. There will be a total floor space of 3360 sq. ft. If we assume the enlargement of this boiler capacity by the substitution of a boiler having a width equal to the 10 boilers and aisles, it would be equivalent to occupying with heating surface the area of 4 aisles and 13 division and side walls. If the latter are assumed to be 18 in. thick, the additional area available for more

boiler would be 940 sq. ft., which at 20.8 sq. ft. of heating surface per sq. ft. of area would amount to 19,552 sq. ft. This would be an enlargement of 39 per cent in the boiler. But on the other hand, if a boiler be operated at two or three times its rated capacity, which is easily done with a good boiler, we have an enlargement of capacity of 200 or 300 per cent.

Assuming uniform quality of fire, the operation of boilers at moderate capacity results in higher heat efficiency than at high capacity. Operation at low capacity requires not only larger investment in boilers, but much greater cost of boiler house structure. Thus high heat efficiency is secured at greatly increased fixed charges, which may be out of proportion to the gain from heat efficiency.

Assuming that the water circulating capacity of a boiler is adequate, the features of the apparatus having greatest influence on capacity are the size of grate and arrangement of gas passages among the heating surface. Conditions of operation having greatest influence on capacity are quality of fire and strength of draft.

In the territory contributory to the Illinois-Indiana-Kentucky coal field, the best coal burning performance of which I am aware is the combustion of 70 lb. of washed nut coal from Illinois per sq. ft. of grate per hour, which was effected on a chain grate stoker, with a chimney height of 175 ft. The latest and most important boiler units to be erected in this territory are at Chicago. These boilers which are horizontal water tube, are of 5550 sq. ft. of heating surface, served by a chain grate stoker of 115 sq. ft. in area, and by a chimney 250 ft. high. Assuming that the performance of 70 lb. of coal burned per sq. ft. be applied to these boilers at an efficiency of 70 per cent, and that the heating power of the coal is 11,896 B.t.u. per lb., we have a h.p. output of 2011, an evaporation of 12.5 lb. of water per sq. ft. of heating surface, and a h.p. from 2.74 sq. ft. of heating surface.

E. D. Dreyfus. When new records are made with a given type of equipment, as brought out in Dr. Jacobus's tests, a comparison with other forms of power-generating apparatus is suggested for consideration when the selection of new installations are determined upon. Building of large boiler and turbine units, has wrought a remarkable improvement in economy, not only from the standpoint of the coal pile, but noticeably so in investment and cost to man and operate.

Viewing first the thermal efficiency, it is of interest to calculate the heat consumption of a steam plant per kilowatt-hour, produced with boilers such as Dr. Jacobus has described with, say, turbines of the

efficiency as reported before the Society.[1] It should be a very conservative estimate to allow that these types of boilers should develop an efficiency of 75 per cent day in and day out with a percentage load considerably under their rating; and this performance includes the stoker auxiliary steam consumption.

Referring to the tests of the City Electric turbine,[1] and selecting the three-fourths load results in order to approximate average running conditions, the B.t.u. consumption per kw-hr. is 17,250 (above discharge water temperature). Including the boiler plant losses, the combined results become 23,000 B.t.u. per kw-hr. In large plants, using steam-driven auxiliaries, with their exhaust heat returned to the boiler feed, the increased consumption they occasion is well within 5 per cent, so that it is safe to say that the plant, over all, should ordinarily run on about 24,000 B.t.u. per kw-hr. generated, netting a plant economy of about 2 lb. of coal per kw-hr., or somewhat better on such high load factors.

Using coal, the producer gas engine layout should be employed for contrast. At three-fourths load, the producer gas engine will not much exceed 11,250 B.t.u. per b.h.p-hr. in regular work, disregarding unusual claims or special tests. With generator losses added (90 per cent), 16,760 B.t.u. per kw-hr., and taking the producer into account, which in commercial practice will develop close to 67½ per cent efficiency, it then becomes about 24,900 B.t.u. per kw-hr. at the switchboard. The auxiliaries must be debited with about 8 per cent of the remaining plant consumption, so that the total results are about 25,640 B.t.u. per kw-hr.

Therefore, the large steam plant, practically considered, is virtually the thermodynamic equal of the gas engine. Oil engines, of course, excel, but together with gas engines and producers they are limited by the size of the units. Hence the same degree of economy of labor and materials cannot be realized with the internal-combustion motor, which leaves them altogether out of the domain of the large power plant.

This comparison no longer holds true for small equipment, the internal-combustion engine remaining quite uniform in efficiency over a wide range, while there is to be observed a very large change in efficiency in the steam plant with size. It is possible that with power only considered and fuel at moderately high prices, the internal-combustion engine may work to an advantage with all factors reckoned.

[1] Trans. Am. Soc. M. E., vol. 32, p. 1251, p. 1267.

Dr. Jacobus's concluding remarks upon the design of boilers capable of being driven to very great overloads, so that the necessity of operating at low loads, and even banking fires, is removed, are of particular economic interest as such a design will eliminate one of the large sources of loss in power plants and increase the handicap on the producer plant which at present does not present similar possibilities.

W. F. M. Goss. The boilers described by Dr. Jacobus constitute an installation of unusual significance not merely because of their size, but because of the excellent distribution of heating surface with reference to grate surface. The form of the flameway is ideal and its ample proportions insure superior results in the combustion of fuel. In this respect the plant is an epoch-making one and Mr. Dow and his company are to be congratulated upon its possession.

The results of the exhaustive tests conducted by Dr. Jacobus are full of interest. Their record of very high efficiency is proof of the thoroughness with which the combustible gases are intermixed, and of the success which has been achieved in reducing to a minimum losses which occur by leakage of air through the setting and in other incidental ways. I am especially interested in the relation between evaporative efficiency and rate of power developed as disclosed by the tests. This relation is one which, in its application to locomotive boilers, I have studied with some care.[1] The decrease in efficiency accompanying increased rates of power as set forth in this paper is not only relatively less than that observed in boilers with which I am familiar, but it is actually very small. This fact suggests the practicability of developing the type of boiler under consideration for work at even higher rates of power than have yet been imposed. It appears probable that by increasing the grate area or by augmenting the rate of combustion upon grates of the present area, this type of Babcock & Wilcox boiler may be made to evaporate 10 or 12 lb. of water per ft. of heating surface per hour with an efficiency which will be well above 70 per cent; that is, the results of these tests suggest the possibility of designing a boiler which may be forced to three times the rate of power now obtained in normal practice, and which at the same time will operate with an efficiency that is practically equal to that obtained from boilers now in common use.

The Author. The manner in which the paper has been received is indeed gratifying, and I wish to thank all who have taken part,

[1] Tests of the Boiler of the Purdue Locomotive, Trans. Am. Soc. M. E., vol. 22. p. 479.

including those who have called attention to what appear to be certain discrepancies. In writing a paper of this sort it is hard to cover all features and often the most useful sort of criticism is that which makes it possible to clear up certain points that might otherwise be questioned later on.

Mr. Rice makes the suggestion that the old practice of rating the capacity of a boiler in horsepower be abandoned and that in place of this the unit be based on the evaporation of 1000 lb. of water into steam per hour. The unit of boiler capacity is now under consideration by the Power Test Committee of the Society, and the matter will be gone into thoroughly before deciding which is the best to adopt.

Mr. Rice believes that means should be employed to protect the boiler by the immediate injection of a large amount of feedwater in case of a sudden stoppage of the turbine unit. The safety valves provide ample protection and as large a margin is provided as in smaller boilers, so that no special precaution need be taken in this respect in operating the boilers.

Mr. De Wolf calls attention to the difference in the drop of pressure through the superheaters in the tests with the Roney and the Taylor stokers. The superheaters were fitted with internal cores and it is possible that some of the cores were left out in the superheaters of the boiler fitted with Roney stokers. This assumption would also account for the lower superheat obtained with the Roney than with the Taylor stokers.

Mr. Pond rightly questions the advisability of using very large boiler units where the feedwater conditions are not favorable. Again where continuity of service is an important feature, the plant must be of a sufficient size to allow at least two of the boilers to be cut out without crippling the capacity, for with one boiler cut out for cleaning or repairs the failure of a single tube in one of the boilers in operation throws two boilers out of service.

Professor Ennis rightly attributes the high efficiencies to the form and dimensions of the furnace. He calls attention to the important effect of carbon monoxide and to the fact that in endeavoring to diminish the excess air to a minimum a point may be reached where a small amount of carbon monoxide will more than offset the gain due to cutting down the air. This has been well appreciated in recent tests. It is a comparatively easy matter to cut down the excess air and obtain, say, 14 per cent of carbon dioxide, but to obtain 14 per cent of carbon dioxide without having any car-

bon monoxide is an entirely different matter. Engineers are often misled into believing there is no carbon monoxide when there may be enough of this gas present to considerably affect the economy. Great care should be taken in preparing the solutions and the gas analyzing apparatus should be tried out to make sure that carbon monoxide will be absorbed before making the regular analyses.

Mr. Parsons says that the furnace meets the requirements he specified many years ago, and congratulates the designers of the boilers on the performance. In addition to the large volume of the furnace an important element leading to the high efficiency is the thorough mingling of the gases in the furnace. Considerably more excess air passes through the fires at the middle line of the furnace at the rear of the two sets of stokers than at the front of the stokers and as the gases at the middle line of the furnace are cooler than the main body of the gases, eddy currents are set up which insure a thorough mixture. The combustible gases from the fresh part of the fuel bed mingle with the gas at the middle part of the furnace which contains excess air, and as the gases are mingled at a high temperature under the action of radiant heat, a complete combustion is effected.

Mr. Esselstyn's data concerning the brickwork are a most valuable contribution in showing how difficulties were met and overcome.

Mr. Bailey questions the accuracy of the coal analyses and the heat value determinations given in the paper. There are variations in some instances that are larger than might be expected, but I know of no similar sets of experiments with which such variations can be compared. It must be remembered that the small samples were obtained separately by different experts and the heat determinations of these different samples made by two different laboratories.

As far as the variation in the heats of combustion of the combustible matter is concerned, it should have been stated in the paper that at times when the large coal storage hoppers began to run low coal of a much different character from that burned on the average was fed to the stokers. The fact that the coal was different was made plain by its appearance and also by its action on the grates. Naturally this might cause a difference in the analyses. Again, the coal was taken from a pile in the yard and there was a very good chance of some other coal being used along with the Red Jacket coal. When it comes to figuring the chemical ash from the test ash and the analysis of the test ash, the difference pointed out is not as large as exists in many of the United States Geological Survey

tests. All of us who have made boiler tests know that it is extremely hard to obtain an accurate sample of the ash in such work. In fact, the only accurate way, where there are a number of very large clinkers, as was the case in the tests in question, would be to pass all of the ash for a test through some sort of a crushing mill and thoroughly mix it and quarter it down. This could not be done, as the total ash for a test varied from, say, 10 to 30 tons, and the samples that were taken represented only approximately the real character of the ash.

Mr. Bailey compares the heating value per pound of combustible with certain values obtained in the work of the United States Geological Survey, and on the basis of the comparison draws the conclusion that the heat values used in calculating the efficiencies and heat balances are about 3½ per cent low. The Red Jacket coal was obtained from the Red Jacket Consolidated Coal & Coke Company, Roanoke, W. Va. This company operates the Lick Fork, Red Jacket, Rutherford and other mines located near Matewan, W. Va., in the southwestern portion of Mingo County. These mines are in the Thacker coal fields, from which samples were collected from southern Mingo County in 1902 and the results published in Vol. 2, 1903, of the West Virginia Geological Survey. The average heat of combustion per pound of combustible obtained in these tests was 15,696 B.t.u. Later on, six samples of coal were collected from this same field and the results published in Vol. 2A, 1908, and Vol. 2, 1911, of the West Virginia Geological Survey. The average heat of combustion per pound of combustible in these tests was 15,094 B.t.u. Lord and Haas also give values for the heat of combustion of Thacker coal in their historic paper on the subject of Calorific Value of Certain Coals, published in 1897 in the Transactions of the American Institute of Mining Engineers. The average heat of combustion per pound of combustible in these tests was about 15,240.

The average of all of the heats of combustion for the Red Jacket coal used in computing the efficiencies given in the paper is about 15,000 B.t.u. per pound of combustible, which approaches the latest determination for coal from the same field made by the West Virginia Geological Survey. The coal comes from a multiple seam containing both "splint" and "gas" coals, hence, there may naturally be a variation in the heat per pound of combustible. Another feature that should be considered in connection with the coal used at Detroit is that it was taken from a pile in the yard where it had been sub-

jected to the weather for some time, and it is a well-known fact that a weathered coal has a less heat of combustion than a freshly mined coal; this in itself would account for a somewhat lower heat value than might otherwise be expected.

The Milburn coal was secured from the Milburn Coal & Coke Company, Keeferton, W. Va. These mines are located near Keeferton, W. Va., in the southwestern portion of Fayette County. The mines tap the Eagle Vein, and analyses were made by the United States Geological Survey on coal from a nearby mine tapping this same vein which show that the coal was of the same general character as the Milburn coal. The average heat value given by the United States Geological Survey is about 15,500 B.t.u. per lb. of combustible. Analyses for this coal are also given in the West Virginia Geological Survey Bulletin 2, 1911, where an average of 15 tests gives 15,631 B.t.u. per lb. of combustible. The average of the figures used in computing the results of the tests of the Detroit boilers is 15,557 B.t.u., which is about a mean of the two figures.

As a further evidence of what he believes are errors in the heating values, Mr. Bailey presents Fig. 21, where the B.t.u. values are plotted against radiation and unaccounted for losses for the first set of heat balances given in Tables 15 and 16, and draws a line on an angle through the scattered points which leads him to conclude that the lower the B.t.u. value the lower the radiation loss. If a similar diagram is constructed for the second set of heat balances given in the same tables, it will be found that no such line can be drawn. The values in the second set are probably more reliable than those in the first, hence Mr. Bailey's conclusion, which is based on the first set, will not hold, as it is not borne out by the second set. The flue gas analyses for the second set of heat balances are more reliable than those used for the first because an average sample for those used in the second set was drawn from six points across the width of the boiler, whereas in the first set the gas samples were drawn from a single point at the middle of the flue and did not represent the average for the entire volume of gases. All this is outlined in the paper but more pains should have been taken to point out the greater accuracy of the flue gas analyses made below the dampers as compared with those made above the dampers.

The foregoing shows that Mr. Bailey's statement that it appears that the average heat value used in calculating the efficiencies and heat balances is about 3.5 per cent low, is unwarranted, and it remains to be shown how far the figures depart from the probable true val-

ues. I have gone on record many times in saying that the sampling of the coal may introduce a considerable error in boiler tests. The magnitude of this error can only be determined by comparing a series of analyses of independent samples in the way that is done in the paper. The estimate made by Mr. Kent that the heat determination results in the present instance are likely to be within 1 per cent of the true value, is in line with my views in the matter, and I further believe that the results for most boiler tests may be wider from the mark than this.

The calorimeters were of the Mahler bomb type. They were calibrated with great care and all precautions known in the art were employed to secure accurate results. The average difference between either one of the calorimeters and the mean value of the heat per pound of combustible, which is the figure to use in comparing the calorimeter results, was about one-third of 1 per cent. The average departure from the mean value per pound of moist coal, which is the figure that directly affects the efficiency results, is about 1 per cent, so that disregarding either one or the other set of calorimeter readings the average efficiency results would be affected by 1 per cent of their value. There are so many chances of error in boiler tests that I feel well satisfied with this limit of error in the heat determinations, and until it can be shown that closer results can be secured in a corrresponding set of tests where the small samples are taken separately and the work is done by different laboratories, I feel safe in saying that the results are as accurate as any I know of.

The Babcock and Wilcox calorimeter has been checked by the U. S. Bureau of Standards napthaline standard and found to give results about one-half of 1 per cent higher than the Government standard. It has also been checked by comparing the heat values with those obtained by other laboratories and found to give substantially the same results, any difference indicating that the results obtained with it are slightly higher than they should be.

Mr. Kent's analysis is of great value in showing the conditions which lead to high efficiency. The efficiencies given in the paper are the combined efficiencies of the boiler and furnace. His formulae indicate that the point of maximum efficiency for the conditions existing in the tests corresponds to the lower capacities and that the falling off in efficiency at the higher ratings is in line with that shown by theory. His careful analysis and the comparison which he makes with other tests show the consistency of the results.

Mr. Kent points out the fact that in many of the tests the carbon monoxide was higher in the flue than it was at the bottom of the

last pass and at the top of the last pass. There are certain features of the flue gas analyses which should be explained. The analyses made in the flue were taken every 15 minutes first on one side of the boiler and then on the other and were used as a guide in running the fires. The samples were taken from a single point in the middle of the flue. In the case of the Roney stokers, we found that the carbon dioxide taken from the middle point of the flue was higher than the average for all of the gas passing through the flue, and that more carbon monoxide was present at the middle of the flue than at other points. The reason for the excess of carbon monoxide in the middle of the flue was probably due to the partition wall which came at the middle of the boiler between the two stokers placed at each side of the boiler, which caused a deficiency of air at this point.

In the case of the tests with the Taylor stokers, there was a leakage of air at the damper boxes where the gases left the boiler, and the gas sampling pipes used at the top of the last pass drew the samples from the side of the flue which was the most affected by the leakage. This reduced the carbon dioxide at the top of the last pass for the Taylor stoker tests below the true average value.

The temperature readings taken at the top of the last pass were also affected by the leakage. The ends of the pyrometer elements were placed in the same position respecting the cross-section of the flue as the sampling nipples for the gas analyzing apparatus. The flue gas analyses taken in connection with the corresponding temperatures are useful for obtaining a heat balance, but when either set of readings is considered by itself, it may be misleading. For example, the flue gas temperatures at the top of the last pass in the tests with the Taylor stoker were less than those taken in the flue.

Mr. Kent asks how the air supply was kept so low in the tests. This was done, as has been stated, by making frequent gas analyses and regulating the air supply on their basis. The results of the analyses were posted on a blackboard in the boiler room and the firemen watched these closely and thickened up the fires when there was too little carbon dioxide and thinned down the fires when carbon monoxide appeared. The furnace conditions secured were certainly remarkable, for to obtain the averages given in the tables it meant that analysis after analysis indicated 14 to 15 per cent of carbon dioxide with no carbon monoxide. There would then follow an analysis containing perhaps as high as 1 per cent of carbon monoxide and the firemen would immediately manipulate the fires to remedy this condition. In most tests only a few of the analyses contained

carbon monoxide, so that the greater part of the time the furnace was running under what might be termed ideal conditions.

Professor Carpenter says he has made many tests of boilers of the same construction as those described in the paper. We suppose that he refers to ordinary Stirling boilers which do not have the same form of furnace as the Detroit boilers. He rightly attributes the high results to good furnace conditions. He asks how the draft was maintained constant at the two sides of the setting. This was done by setting the dampers once for all so that in regulating the amount of opening the dampers on each side would be opened the same amount. The constancy of the draft on the two sides was checked by means of a draft gage.

Mr. Billings calls attention to the variations in the heat of combustion obtained in the two laboratories, and what I have said in reply to Mr. Bailey answers the points that he raises. The main difference between the two laboratories was due to the samples and until we have a correspondingly long series of tests where the sampling and calorimeter work by two laboratories is checked one against the other, and where every result is reported even though it may be out of line, we cannot say whether or not the results vary more than they should.

Mr. Thomas calls attention to the action of the flames in passing upward along the boiler tubes and thus imparting a maximum amount of heat to the tubes. Another feature that may be interesting is the fact that any leakage in the lower part of the baffles at the rear of the tubes which come next to the furnace will be from the cooler gases into the furnace instead of from the furnace outward. This is caused by the flue action of the high furnace which produces a greater suction at the bottom of the baffles on the furnace side than exists at the bottom of the second pass. The action is important as it eliminates any tendency of the hot gases to short circuit at this point. Again, baffle leakage from the furnace to a cooler portion of the boiler often induces a loss through carbon monoxide on account of the chilling off of the partially burned gases, and with the Detroit boilers such an action cannot take place.

Mr. Bement is right in his views respecting the low amount of radiation in well set boilers. In some tests made on the Pacific Coast with oil fuel, we found that the steam pressure could be maintained indefinitely by burning 2 per cent of the oil required to run the boilers at rating. We also found by careful heat balances that the radiation and unaccounted for losses amounted to about 2 per cent.

Naturally, when simply maintaining the steam pressure there would be less radiation than when running the boilers at rating on account of the lower temperature of the boiler walls, but a portion of the heat generated by the oil burned to maintain the steam pressure escaped to the stack which tended to counterbalance the effect of the lesser radiation. Mr. Bement also says that water may leave a boiler with the steam without being detected. We made careful calorimeter observations which proved that the saturated steam entering the superheaters was dry. We further went to the precaution of measuring the superheat in the steam after it had passed through a "Y" where the steam from one side of the boiler was mingled with that from the other. The temperature of the steam after being mingled at the "Y" was practically the same as that obtained by averaging the temperatures of the steam discharged from the two superheaters and both figures are given in the tables. The readings of the thermometers used in measuring the temperature of the superheated steam were corrected for the effect of cooling the mercury in the exposed part of the stems and were reduced to equivalent temperatures by an air thermometer.

Mr. Dreyfus' conclusion respecting the efficiency of a steam plant attainable through using the Detroit boilers as compared with that of a producer gas engine plant is indeed interesting, since he shows that one is the equal of the other thermodynamically and that the steam plant excels when the cost of labor, etc., is considered.

Dr. Goss attributes the high efficiencies to the excellent distribution of the heating surface with reference to the grate surface and the form of the flameway. His remarks touch the keynote of the subject, as the combined arrangement certainly forms a distinct advance in boiler furnace construction.

No. 1329

STRAIN MEASUREMENTS OF SOME STEAM BOILERS UNDER HYDROSTATIC PRESSURE

By James E. Howard,[1] Washington, D. C.

Non-Member

The object of these tests is to ascertain the condition of the metal of the shell and other parts of two horizontal tubular steam boilers which had been in use for a term of service of unusual length; and in addition thereto to acquire information on constructive details by means of measured strains.

2 The boilers were contributed for investigative purposes by the treasurer of the Kendall Manufacturing Company, Providence, R. I., the late Nicholas Sheldon, Esq. They were of early manufacture and from their remarkable history and present condition were of special value for these tests. They were made by the Whittier Machine Company, Boston, Mass., using "Benzon" brand of steel, and were put into service March 1881. They were in continuous service for a period of 27 years, during which time, as Mr. Sheldon wrote, "no repairs were required; in fact, not one cent has been spent upon them."

3 They consisted of five course boilers, two sheets to a course, having the following general dimensions:

Diameter, in.	72
Length over dry sheet, ft.	16
Thickness of shell, in.	$\frac{3}{8}$
Thickness of heads, in.	$\frac{1}{2}$
Number of tubes.	140
Diameter of tubes, in.	3
Length of tubes, ft.	15
Diameter of dome, ft.	$2\frac{1}{2}$

Longitudinal seams, double-riveted lap joints, $\frac{3}{4}$-in. rivets, 2-in. pitch, punched holes, rows $2\frac{1}{8}$ in. apart, rivets staggered.
Girth seam, $\frac{3}{4}$-in. rivets, $2\frac{1}{8}$-in. pitch.
Heads stayed, each, with 14 braces.
Cast-iron manhole frames and safety-valve nozzle.
Supported by lugs, three on a side.
The feedwater came from the Pawtucket River.

[1] Engineer-Physicist, Bureau of Standards, Washington, D. C.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

4 The hydrostatic tests were made at the W. H. Hick's Boiler Works, Providence. Mr. Francis B. Allen, Vice-President of the Hartford Steam Boiler Inspection & Insurance Company, assisted and advised with the writer in conducting them. The boilers will be designated by the numbers 4084 and 4092 under which they were carried on the books of the Hartford company.

5 The tests began with strain measurements upon different parts of the boilers as they were subjected to successive increments of hydrostatic pressures. The results of this portion of the inquiry are now available and herewith presented. Much remains to be done in the other direction of testing, pertaining to the physical properties of the materials.

6 Measurements of the deformations of engineering structures, whether steam boilers, bridges or buildings, may be expected to develop information of a kind not attainable in the tests of the component parts of those structures. A comparatively new field of inquiry is presented in the tests of structures over the tests of the materials thereof. The effects of combined stresses may readily be studied in this manner.

7 No more simple type of boiler could be chosen than the plain horizontal, tubular boiler of these tests, yet it will be seen from the results that complexity of strains and stresses are found in most parts of the shell. In comparatively few places are tangential strains displayed corresponding in magnitude to those which would be expected in a thin cylindrical shell subjected to a given interior pressure.

8 Ascertaining the deformations by the method of measured strains, locally determined, consists of establishing gaged lengths on different parts of the boiler and then measuring them initially and at intervals as the hydrostatic pressures are successively applied and released.

9 Gaged lengths of 10 in. each were used in the examination of these boilers. Their extremities were defined by small drilled and reamed holes. The holes are about 0.05 in. in diameter by, say, 0.10 in. deep, and reamed to a conical shape. The angle of the reamer is 65 deg., and the distance across the hole at the surface of the shell sheet about 0.08 in.

10 Such holes carefully made, in metal surfaces, are capable of centering with considerable precision the contact points of the micrometer strain gage. The strain gage is used as a transfer instrument to compare the gaged lengths on the work with a corresponding length on a standard reference bar.

11 Fig. 1 shows the 10-in. strain gage used on these tests. It consists of two principal parts, an outer tube and an inner stem, which are telescopic, working on ball bearings. Each part carries a conical contact point for centering the instrument on the reference bar and on the work. A screw micrometer measures the length of the instrument when in position.

12 The conical contact points of the strain gage have an angle of 55 deg. This difference of 10 deg. between the reamed hole in the boiler shell and the points of the gage secures contact at a short distance below the surface of the shell. Ordinarily the reference holes are safe against accidental injury, due to their position.

13 As to the degree of precision attained with the strain gage, in the hands of skilled manipulators and under favorable conditions, such as were experienced with these boilers, it is believed the readings are generally reliable to one ten-thousandth of an inch. This strain corresponds to a stress of 300 lb. per sq. in. on a 10-in. gaged length,

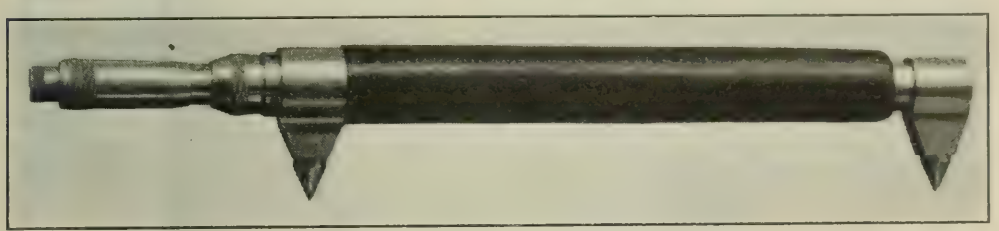


FIG. 1 10-IN. STRAIN GAGE

using a modulus of elasticity of 30,000,000 lb. Fig. 2 shows boiler No. 4084. Both boilers were of the same dimensions except at the dry sheets. When on their settings, boiler No. 4084 was on the right, boiler No. 4092 on the left side. This view shows the locations of some of the gaged lengths which were established on this boiler, taken in both tangential and longitudinal directions.

14 A more comprehensive series of lengths was established on boiler No. 4092, and the general discussion of the results of the strain measurements will be given in connection with the test of that boiler.

15 In the test of No. 4084 greater strains were displayed in the vicinity of the dome and the manhole frame than at other parts of the shell. This resulted, as would clearly be expected, in the early failure of the boiler at those places.

16 Actual rupture of the dome was not accomplished, but leakage along its single-riveted longitudinal seam became so great at 266 lb. pressure that it was necessary to remove the dome and patch the shell in order to reach higher pressures with the pump available.

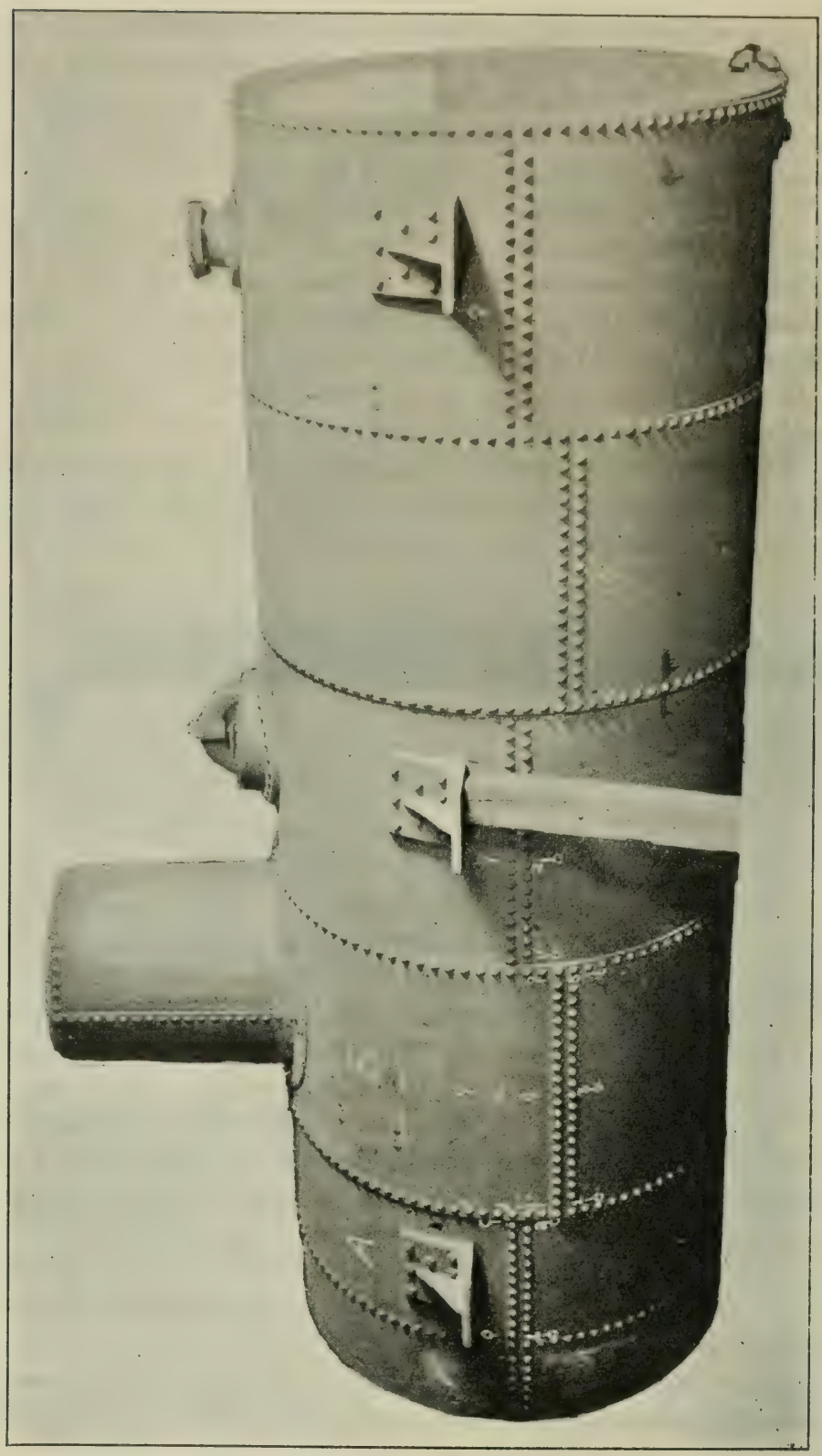


FIG. 2 STEAM BOILER NO. 4084 BEFORE TESTING

17 At 270 lb. pressure the cast-iron manhole frame fractured across the middle of its length. Another patch was then put on the shell covering the manhole.

18 The test was again resumed when at 295 lb. pressure the rupture of three braces of the front head occurred. The test was then discontinued and the boiler dismantled.

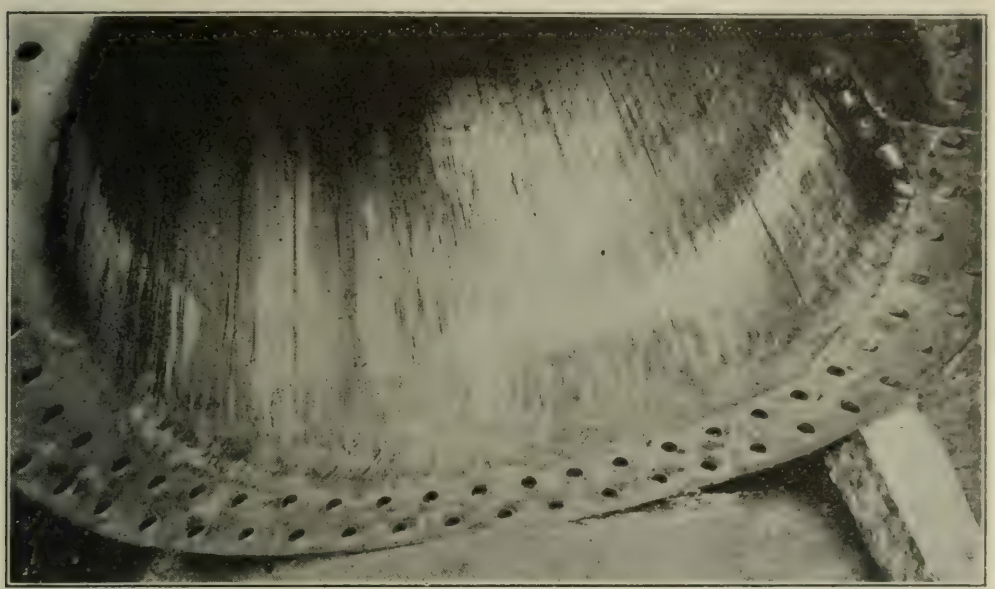


FIG. 3 INTERIOR OF DOME SHOWING LINES ALONG WHICH SCALE WAS DISTURBED AFTER PRESSURE OF 266 LB. ON SHELL

19 The strain measurements made in the test of No. 4084 were of the same general order as those subsequently made in the test of the second boiler and the results were, for the most part, quite similar.

20 A feature, however, in the test of the first was absent or obscure

TABLE 1 TANGENTIAL EXTENSIONS OF THE SEAMS, BOILER NO. 4084

Pressures	Course			
	B	*C*	*D*	*E*
210	0.0166	0.0121	0.0099	0.0084
240	0.0241	0.0171	0.0138	0.0121
270	0.0341	0.0241	0.0212	0.0187

in that of the second. There was a progressive difference in the extensibility taken across the longitudinal seams of the several courses in passing from the front to the rear end of the boiler.

21 The tangential extensions of the seams, at the middle of their lengths, were as shown in Table 1.

22 While these seams were not directly exposed to the heated gases over the grate, nevertheless it seems probable that a wider range of thermal conditions prevailed in the vicinity of the seams at the front end over those at the rear end of the boiler. If such was the case it would aid in explaining the greater slip of the forward seams.

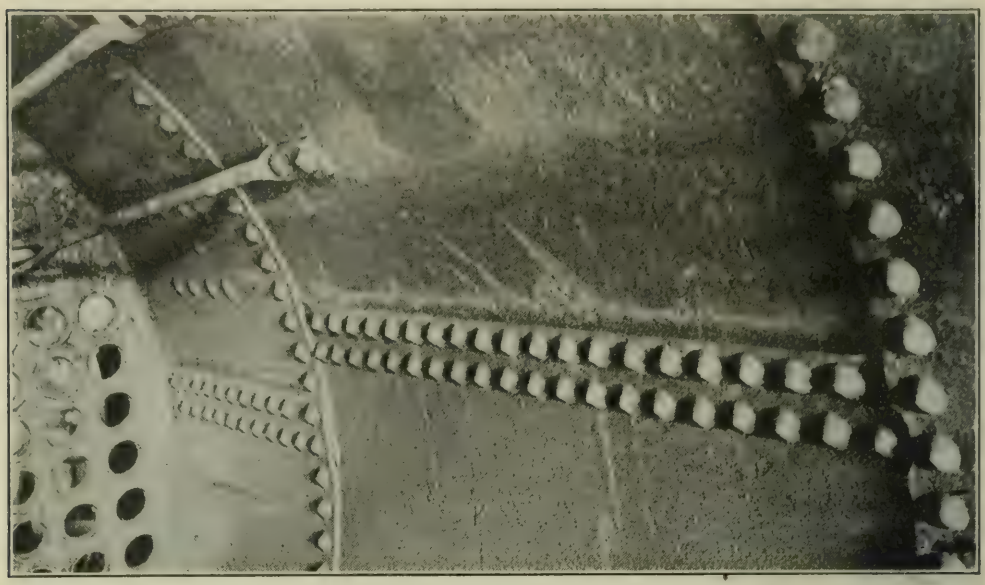


FIG. 4 INTERIOR VIEW OF BOILER NO. 4084 AFTER 295 LB. PRESSURE SHOWING SCALE DISTURBED IN VICINITY OF THE LONGITUDINAL SEAMS

23 Hydrostatic pressures on the exterior surfaces of the tubes necessarily extend them in length. The amount of the extension appears to depend upon their position with reference to their proximity to the shell. Tubes adjacent to the shell extended less than those at the middle of the rows, a restraining influence from the shell appearing to affect the outer ones.

24 The results in Table 2 were obtained by measuring the tubes over their full length.

25 Practically no leakage occurred about the tubes throughout the test of this boiler. A slight leakage took place at two tubes at 120 lb. pressure, but soon ceased and was not renewed during the remainder of the test. The girth seams remained tight up to 210 lb. pressure, and then showed only small leaks which were not materially increased under the higher pressures.

26 Leakage at the longitudinal seams began at 120 lb. pressure and increased as higher pressures were applied. The leakage became general at these seams with 180 lb. pressure on the boiler, but at

this time the slip of the joints had become a pronounced feature of the case, which necessarily disturbed the calking.

TABLE 2 EXTENSION OF TUBES, BOILER NO. 4084

Pressures	Third Row		Seventh Row	
	Next Shell	Middle of Row	Next Shell	Middle of Row
210	0.0110	0.0162	0.0077	0.0167
240	0.0128	0.0184	0.0086	0.0189
270	0.0142	0.0210	0.0099	0.0210

27 Upon removal of the dome, evidence of overstraining was found at its base next the flanged portion. The scale had been disturbed on the inside on the line and in the vicinity of the upper element of the boiler, as shown in Fig. 3. Near the flange the scale was disturbed in oblique, shearing directions, which changed to longitudinal and then tangential directions a little farther up the dome.

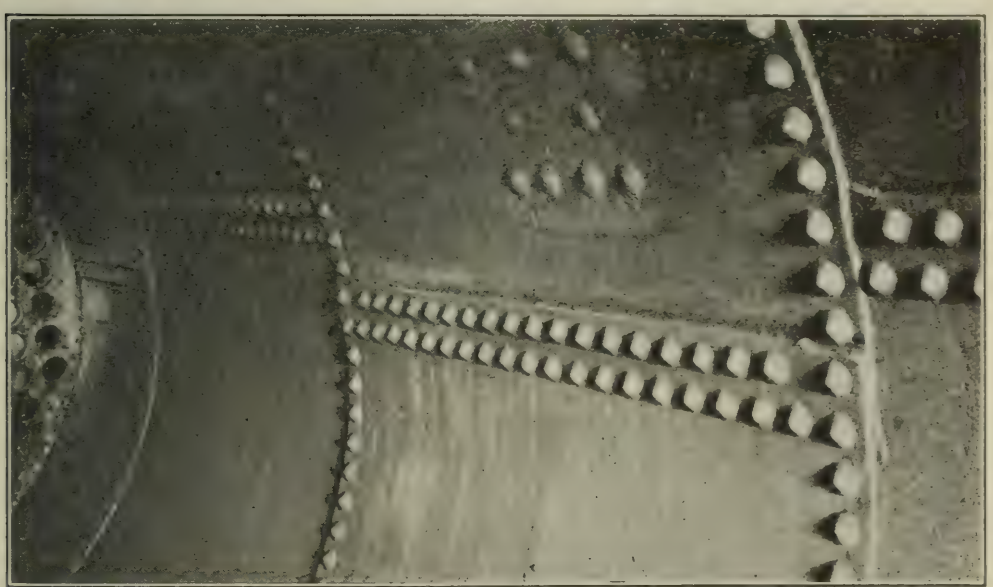


Fig. 5 Interior View of Boiler No. 4084 after 295 Lb. Pressure Showing Scale Disturbed in Vicinity of the Longitudinal Seam and under Supporting Lug

28 Figs. 4 and 5 are interior views of the boiler illustrating the manner in which the scale was disturbed during the test in the vicinity of the longitudinal seams and under one of the lugs.

29 Struts were used under the middle lugs and supported a part of the weight of the boiler during the test. They probably intensi-

fied the stresses in the shell in that vicinity. The interior surface of the shell and also the heads were found in good condition. Fig. 6 shows the appearance of the inside of the rear head.

30 A series of six photographs, Figs. 7 to 12 inclusive, shows the appearance of the exterior surfaces of the tubes. The system of lettering and numbering the horizontal and the vertical rows is indicated in Fig. 6. The layout of the feed pipes appears on Fig. 13.

31 Tubes in the horizontal row marked *D*, the fourth in the boiler from the top, Fig. 7, had a deposit on the rear third of their length, and also a slight deposit on the front ends. On some of the lower rows the deposit was thicker, as Figs. 8–12 indicate. In general the deposit was greatest in the lower rows of tubes and on those farthest from the shell, being confined chiefly to the rear quarter or half of their lengths. The lower rows of tubes, at the front end of the boiler, had a deposit on them. The surfaces of the upper row and the side rows were clean without deposit.

32 Material collected from the bottom of the boiler at the front and rear ends had the following approximate chemical composition:

Deposit from front end of boiler:	Per Cent
Loss on ignition	26.00
SO_3	4.10
Silica	21.60
$Fe_2O_3+Al_2O_3$	25.40
Lime	5.30
Magnesia	13.10
Copper oxide (about)	0.25
CO_2	slight amount
Chlorides	trace

Deposit from rear end of boiler:	Per Cent
Loss on ignition	23.85
SO_3	trace
Silica	27.60
$Fe_2O_3+Al_2O_3$	24.80
Lime	9.60
Magnesia	12.70
Copper oxide (about)	0.25
Chlorides	trace
Carbonates	small amount

33 Prior to testing boiler No. 4092, it was stripped of its dome and manhole frame and the shell patched at those places. The heads were strengthened by means of six $1\frac{1}{4}$-in. braces extending from head

to head. The cast-iron safety-valve nozzle was allowed to remain in place, but was eventually replaced by a soft patch, after 300 lb. pressure had been applied and released. The distortion of the shell under the flange of the nozzle caused leaks impracticable to calk.

34 Fig. 14 shows the boiler when about ready for the hydrostatic test. It was supported on two wooden shoes sawed to fit the curvature of the shell, in lieu of the blocking shown in the illustration.

35 The gaged lengths which were established on the right side

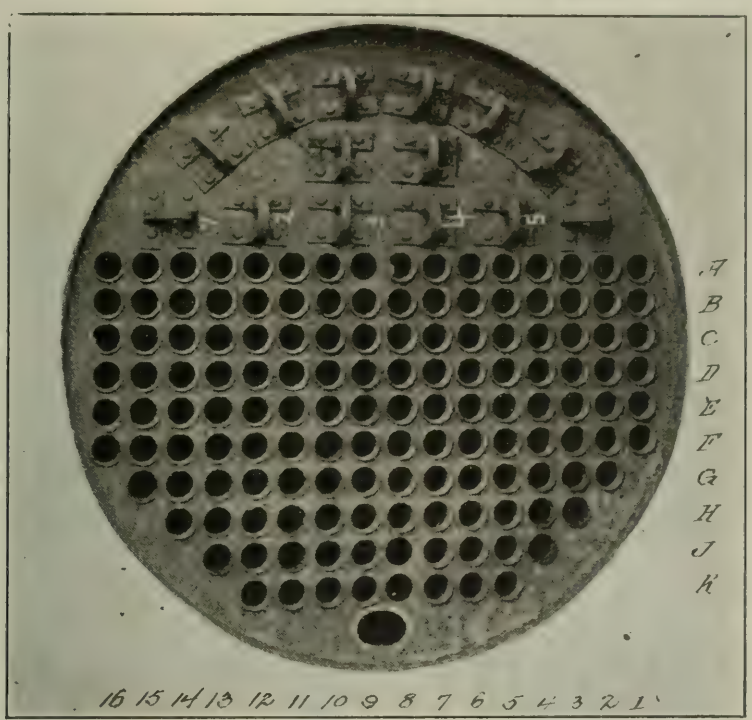


FIG. 6 INSIDE OF REAR HEAD, BOILER NO. 4084, AFTER 295 LB. PRESSURE. LETTERS AND FIGURES REFER TO MARKS SHOWING LOCATION OF TUBES IN SUBSEQUENT PHOTOGRAPHS

of the boiler are shown in this figure and practically stand for those on the left side, while additional ones were laid off on the top. Figs. 15, 16 and 17 show, however, the different gaged length on each side and the top. Additional gaged lengths were laid off and measured which are not shown on these diagrams; their results confirm those which will be referred to. There were 165 gaged lengths used in the principal series of observations, on which some 3300 readings were taken.

36 The general results of the strain measurements have been plotted on a series of ten diagrams, Figs. 18 to 27, inclusive. For the

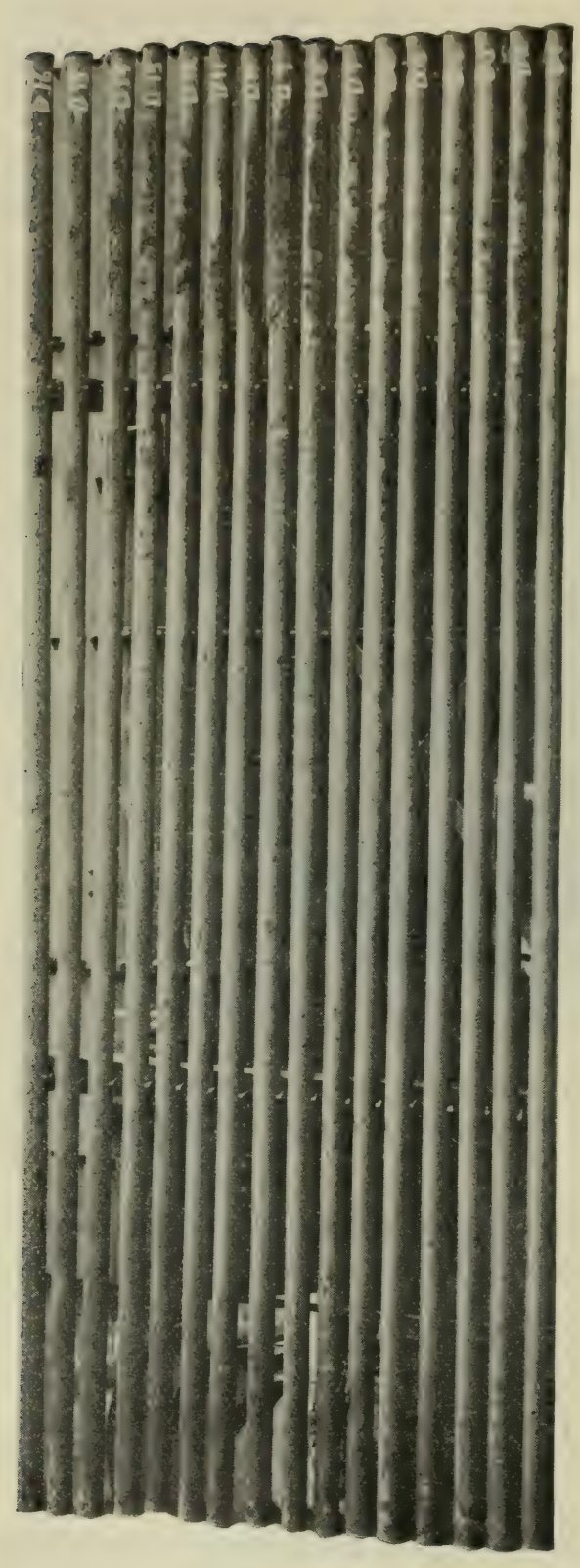


FIG. 7 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. HORIZONTAL ROW *D*, FOURTH FROM TOP

FIG. 8 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. VERTICAL ROW, FIRST ON RIGHT SIDE OF BOILER

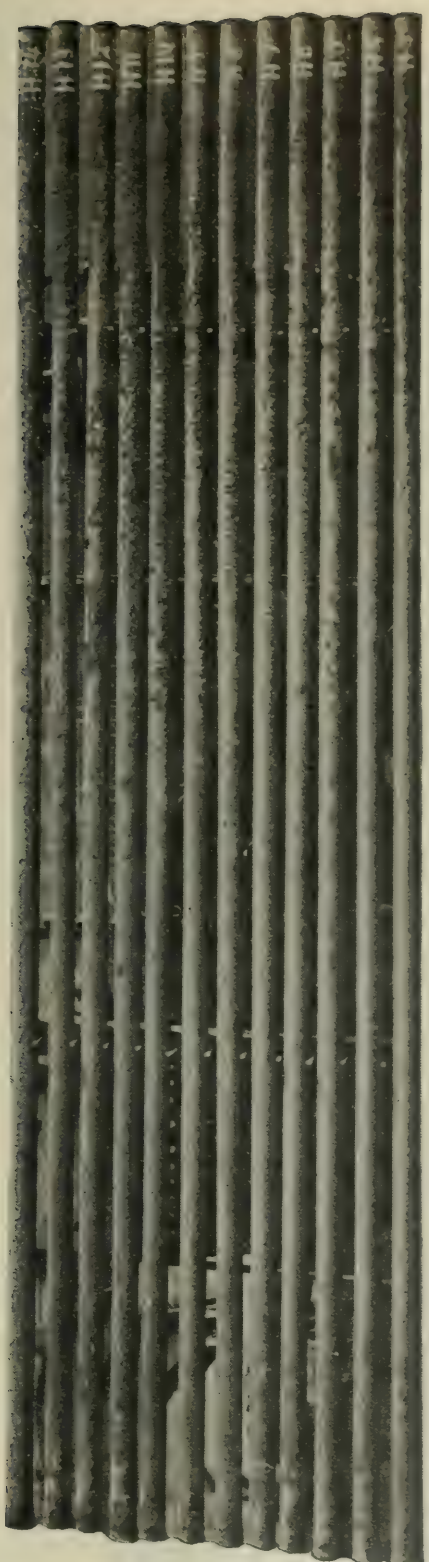


FIG. 9 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. HORIZONTAL ROW *H*, EIGHTH FROM TOP

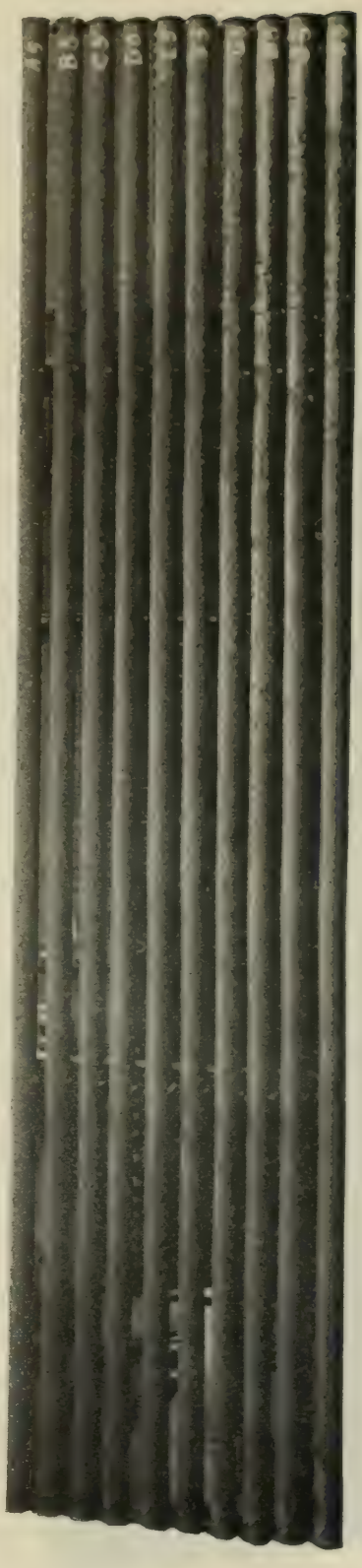


FIG. 10 APPEARANCE OF TUBES, BOILER NO. 4084 AFTER DISMANTLING. VERTICAL ROW, FIFTH FROM RIGHT SIDE OF BOILER

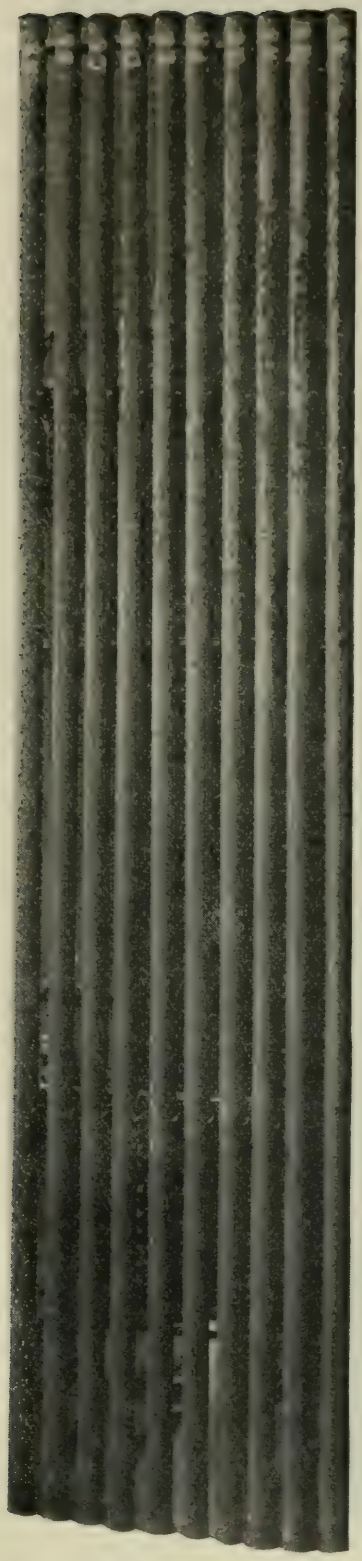


FIG. 11 APPEARANCE OF TUBES, BOILER NO. 4084, AFTER DISMANTLING. VERTICAL ROW, EIGHTH FROM RIGHT SIDE OF BOILER

FIG. 12 APPEARANCE OF TUBES, BOILER NO. 4048, AFTER DISMANTLING. VERTICAL ROW, TWELFTH FROM RIGHT SIDE OF BOILER

purpose of furnishing a convenient basis of comparison, in judging the behavior of the boiler at different parts and under different pressures, heavy lines have been drawn on each diagram which indicate strains corresponding to those which would be displayed by the

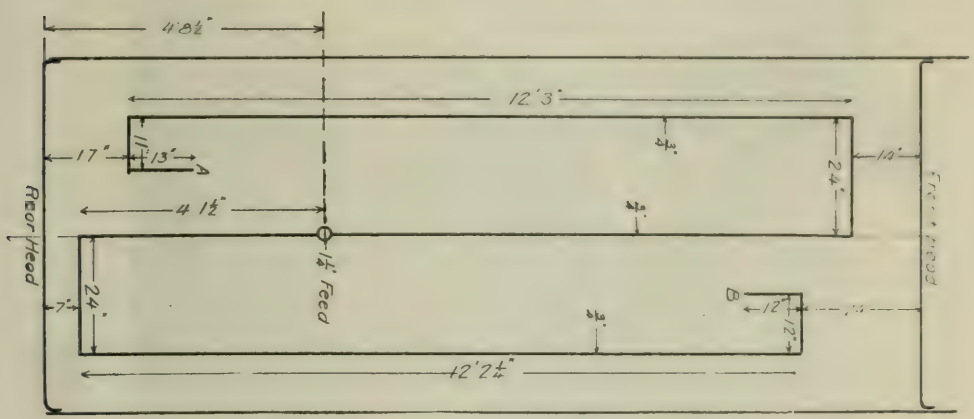


FIG. 13 LAYOUT OF FEED-PIPES
Pipe *A* Discharged directly over Tubes of Row No. 5 at Rear End
Pipe *B* Discharged directly over Tubes of Row No. 12 at Front End

TABLE 3 COMPUTED STRESSES ON THE SHELL SHEETS, BOILER NO. 4092

Boiler Pressure, Lb. per Sq. In.	Stress on ¾-In. Shell, Lb. per Sq. In.	Strain on Gaged Length of 10 In.
30	2880	0.0010
60	5760	0.0019
90	8640	0.0029
120	11520	0.0038
150	14400	0.0048
180	17280	0.0058
210	20160	0.0067
240	23040	0.0077
270	25920	0.0086
300	28800	0.0096

sheets under direct tensile stresses, using a modulus of elasticity of 30,000,000 lb. per sq. in. Plotted curves, which are steeper than the modulus of elasticity reference line, indicate places on the boiler having greater rigidity than normal to the plain sheets; while flatter curves indicate greater extensibility than pertains to the plain metal.

37 Tangential rigidity above the normal was displayed in the vicinity of the girth seams, while the gaged lengths taken across the longitudinal seams at the middle of the length of the courses showed a much lower degree of rigidity than common to the plain sheet. Zones of greater extension than normal were also found in the vicinity of the manhole and dome.

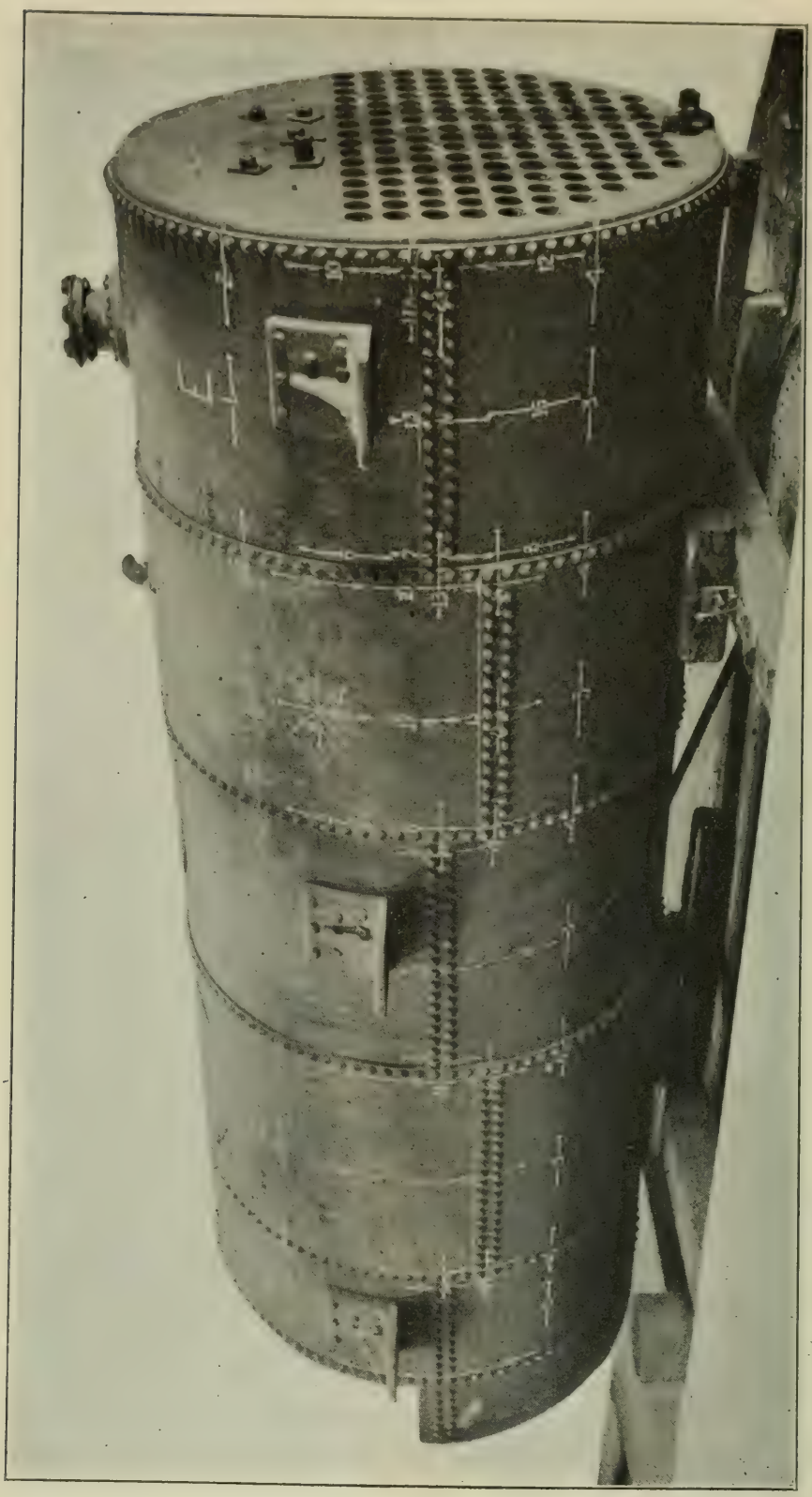


FIG. 14 STEAM BOILER, NO. 4092, DOME AND MANHOLE FRAME REMOVED PREPARATORY TO TESTING. HEADS STRENGTHENED WITH 6 THROUGH BRACES

38 Flattening of the curves representing the solid sheets necessarily accompanied those pressures which caused a tensile stress on the shell in excess of its elastic limit.

39 Table 3 shows the computed stresses on the shell sheets, considering only tangential stresses as acting, and the strains which should

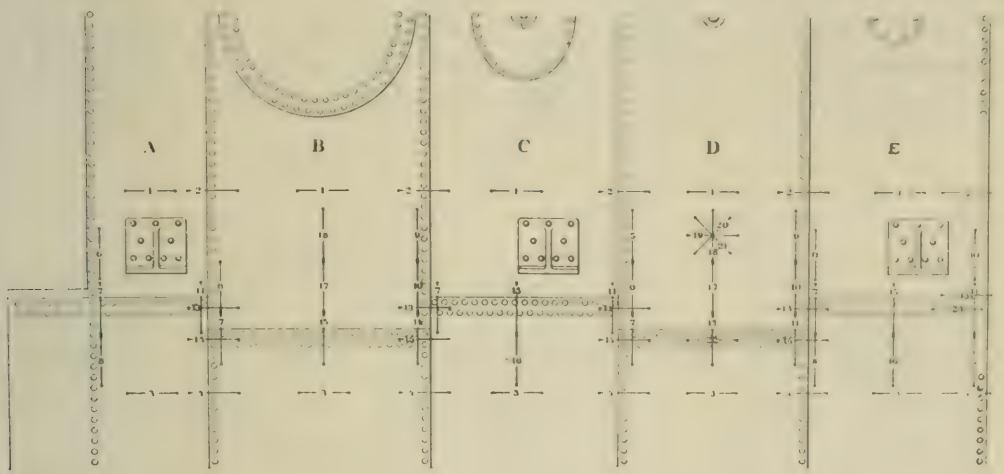


FIG. 15 DIAGRAM SHOWING LOCATION OF GAGED LENGTHS, 10 IN. EACH, LAID OFF ON RIGHT SIDE ON BOILER NO. 4092

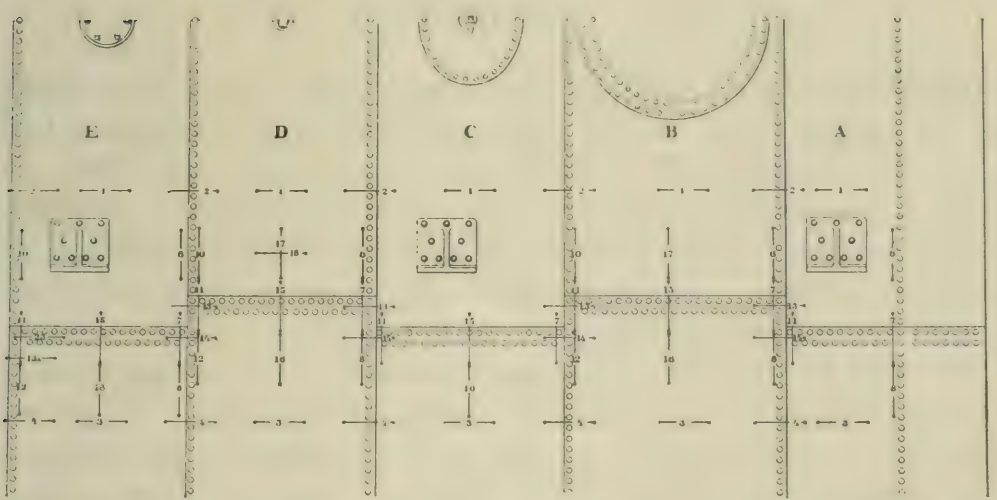


FIG. 16 DIAGRAM SHOWING LOCATION OF GAGED LENGTHS, 10 IN. EACH, LAID OFF ON LEFT SIDE OF BOILER NO. 4092

be developed on a gaged length of 10 in., using a modulus of elasticity of 30,000,000 lb., the interior diameter of the boiler being 72 in.

40 Referring now to the plotted results, Fig. 18 shows the tangential extensions of sheets *C* and *D* on the right and left sides of the

boiler respectively, taken at the middle of the lengths of the courses. Gaged length *D*-18, on the right side of the boiler, was located above the longitudinal seam, while *C*-16, on the right side, was located below the longitudinal seam.

41 The tangential extensions of each of these gaged lengths closely follow the modulus of elasticity comparison curve. The departure of *D*-18 from this line does not exceed 0.0002 in. at any pressure, and coincides with it at several pressures. The extensions displayed by the gaged lengths on the opposite side of the boiler agreed fairly well

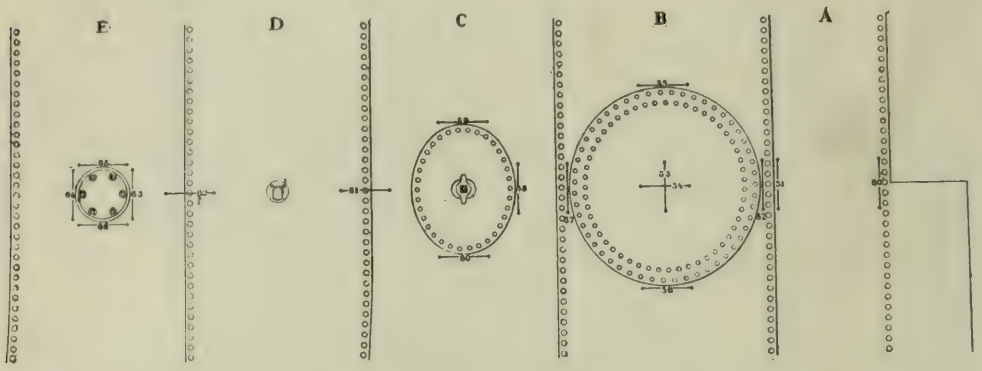


FIG. 17 DIAGRAM SHOWING LOCATION OF GAGED LENGTHS, 10 IN. EACH, LAID OFF ON TOP OF BOILER NO. 4092

with the modulus of elasticity reference line also, but not so closely as the results found on the right side, while rapid extension took place one increment of pressure earlier than on the right side. So close a correspondence between the measured and the computed strains as shown on this diagram did not, however, characterize many places on the shell. Commonly there were modifying influences which disturbed the normal display of elastic extensions of the metal.

42 Fig. 19 shows that the tangential strains near the girth seams, *D*-5 and *D*-9, were less than at the middle of the course. In general this behavior was shown in the other courses, but an exception was found on the left side of the boiler in course *B* where substantially the same rigidity was displayed at the middle as at the edges of the course.

43 In the third group of curves on this diagram, however, the extension of *D*-16 taken at the middle of the course is seen to be greater than at *D*-8 and *D*-12, curves representing the edges.

44 There is a marked difference in the tangential extension of the two edges of the end courses of the boiler, due to the influence of the

heads in supporting the shells. Fig. 20 shows the greater rigidity of gaged lengths *E*-12 which are taken nearly over the rear head, than at the other places, which were measured on this course.

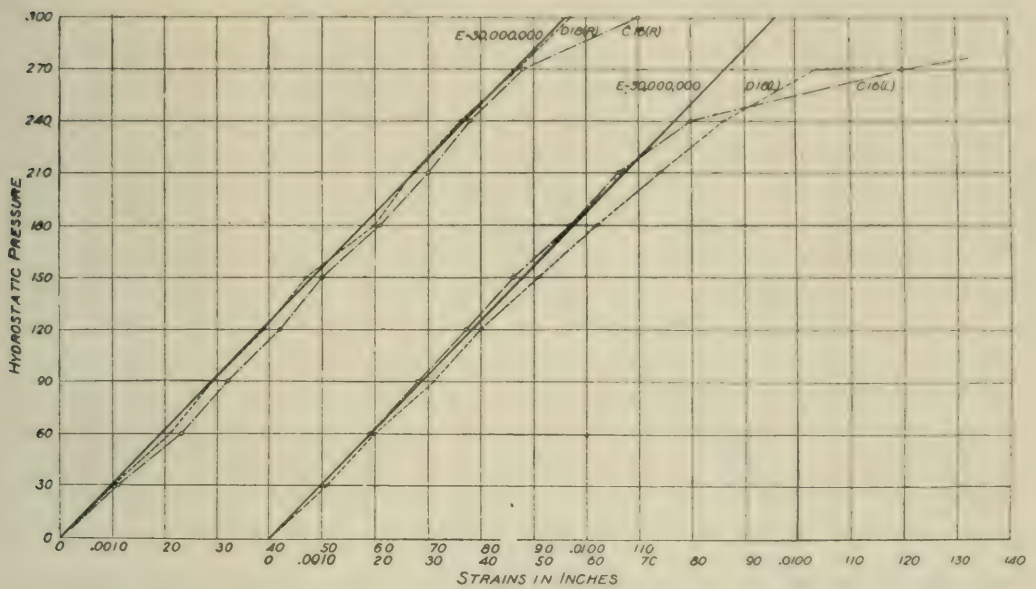


FIG. 18 CURVES OF TANGENTIAL EXTENSION, SOLID SHEETS, AT MIDDLE OF LENGTH OF COURSES *C* AND *D*, RIGHT AND LEFT SIDES OF BOILER

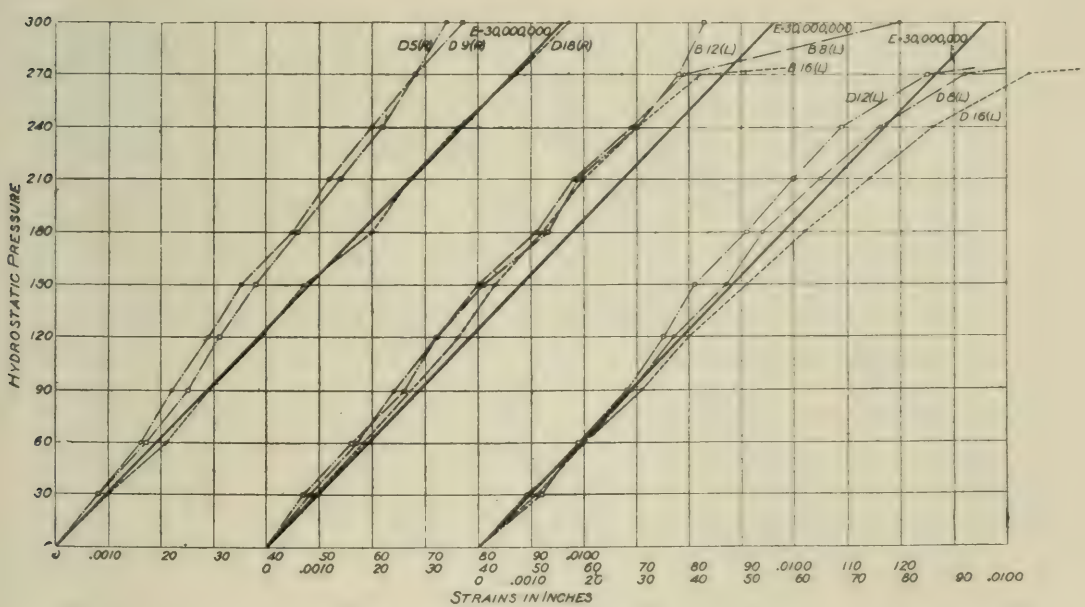


FIG. 19 CURVES OF TANGENTIAL EXTENSION, SOLID SHEETS, NEAR GIRTH SEAMS AND AT MIDDLE OF LENGTHS OF COURSES

45 In regard to the top of the boiler there were many disturbing factors present, as indicated by Fig. 21. The first course is a short

one with the front head to stiffen one edge. Then came the dome in the original construction on course *B*, which was patched, and the

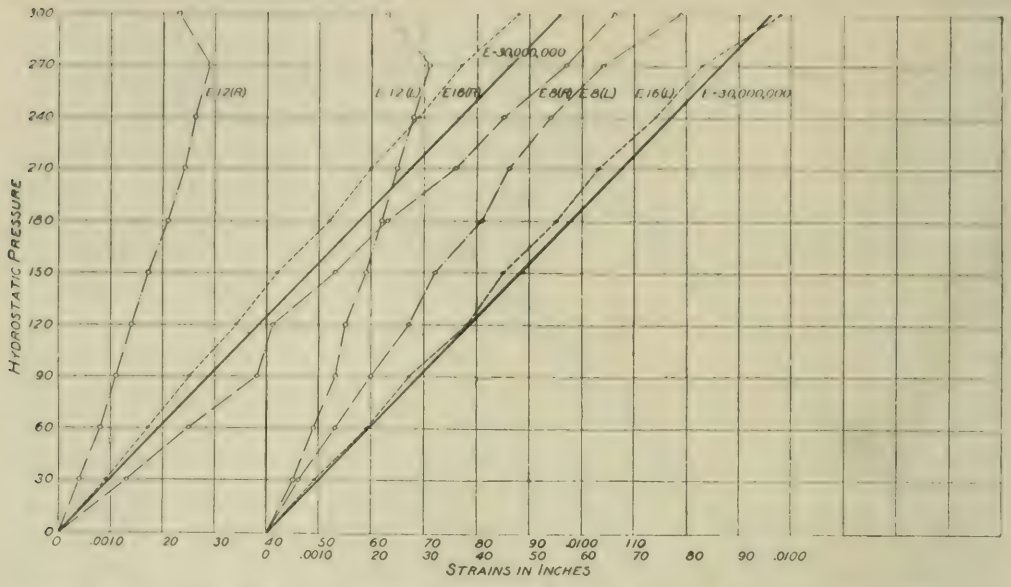


FIG. 20 CURVES OF TANGENTIAL EXTENSION, SOLID SHEETS, END COURSE *E*, NEAR REAR HEAD, GIRTH SEAM, AND MIDDLE OF LENGTH OF COURSE

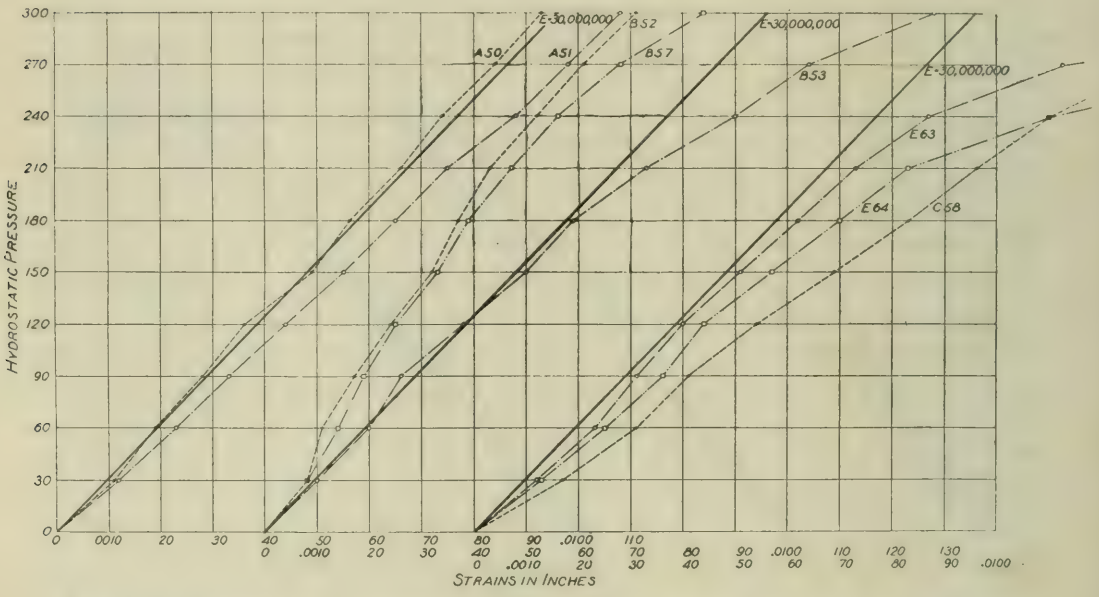


FIG. 21 CURVES OF TANGENTIAL EXTENSION, TOP OF BOILER, NEAR FRONT HEAD, GIRTH SEAMS, DOME AND MANHOLE PATCHES, AND SAFETY-VALVE NOZZLE

patch double-riveted, using the rivet holes that were made for securing the flange of the dome to the shell. Course *C* had the manhole

patch, a single riveted one, using the holes made for securing the manhole frame to the shell. In course *D* was found the feedwater

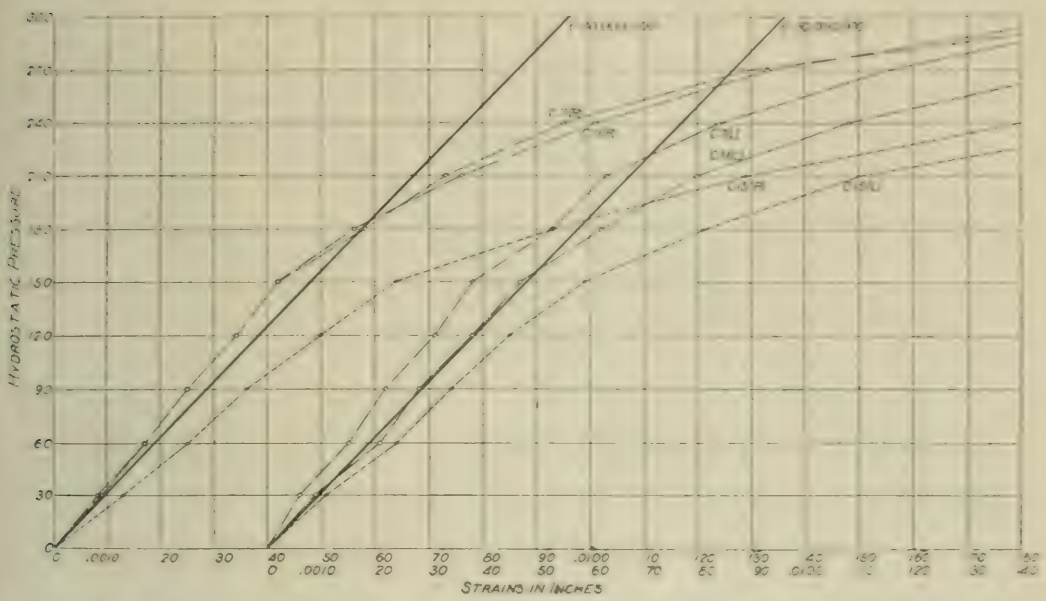


FIG. 22 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS, AT MIDDLE OF LENGTH OF COURSE *C*, AND AT EDGES, RIGHT AND LEFT SIDES OF BOILER

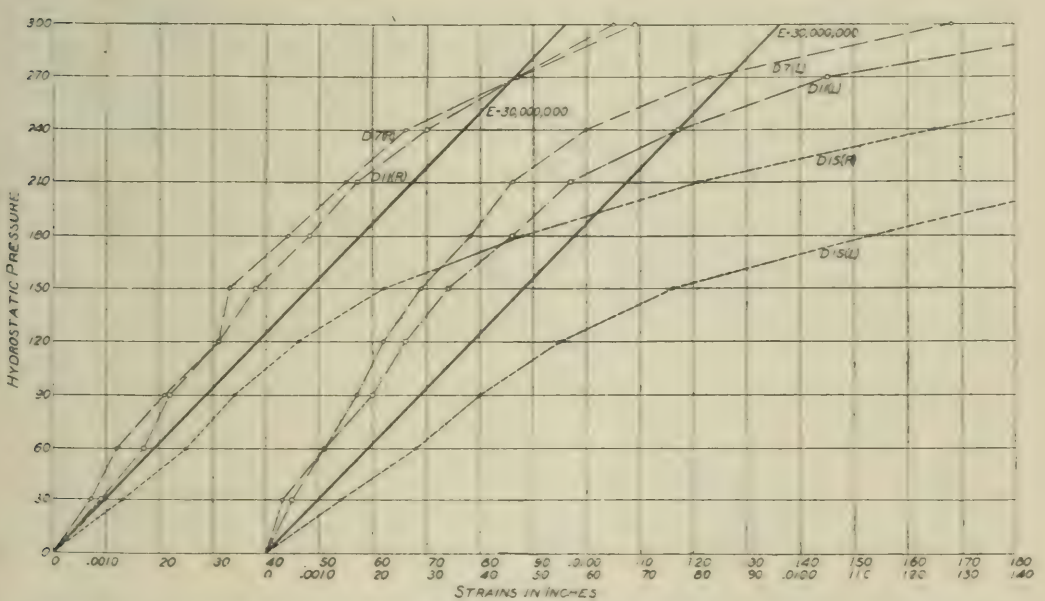


FIG. 23 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS, AT MIDDLE OF LENGTH OF COURSE *D*, AND AT EDGES, RIGHT AND LEFT SIDES OF BOILER

connection which probably did not have much influence on the behavior of this course while under pressure. Course *E* had the cast-iron

safety-valve nozzle riveted to it, which did seem to have an influence on the tangential extension of the steel, permitting greater extension than normal.

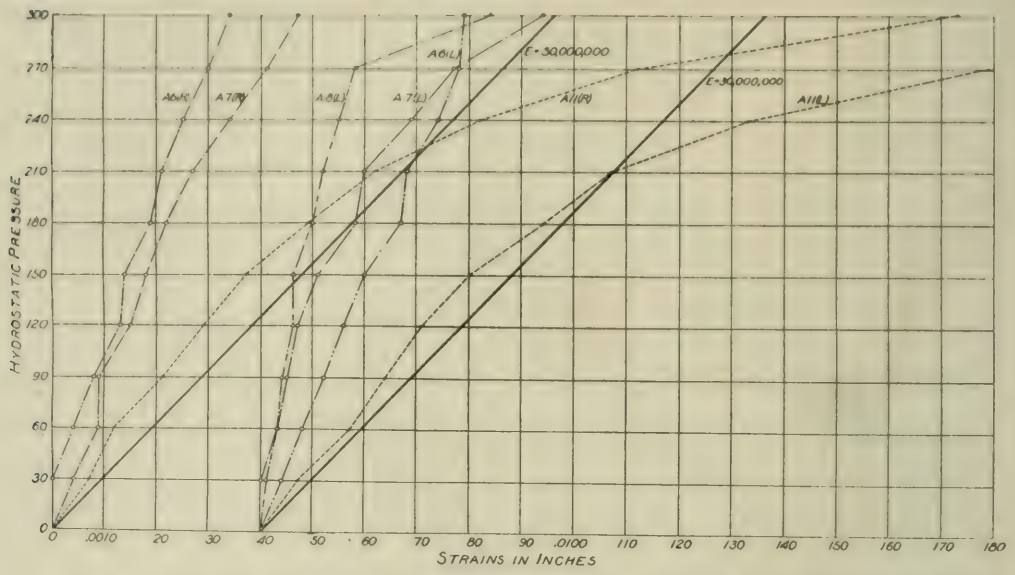


FIG. 24 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS AND SOLID SHEETS, END COURSE *A*; EXTENSIONS NEAR FRONT AND GIRTH SEAM

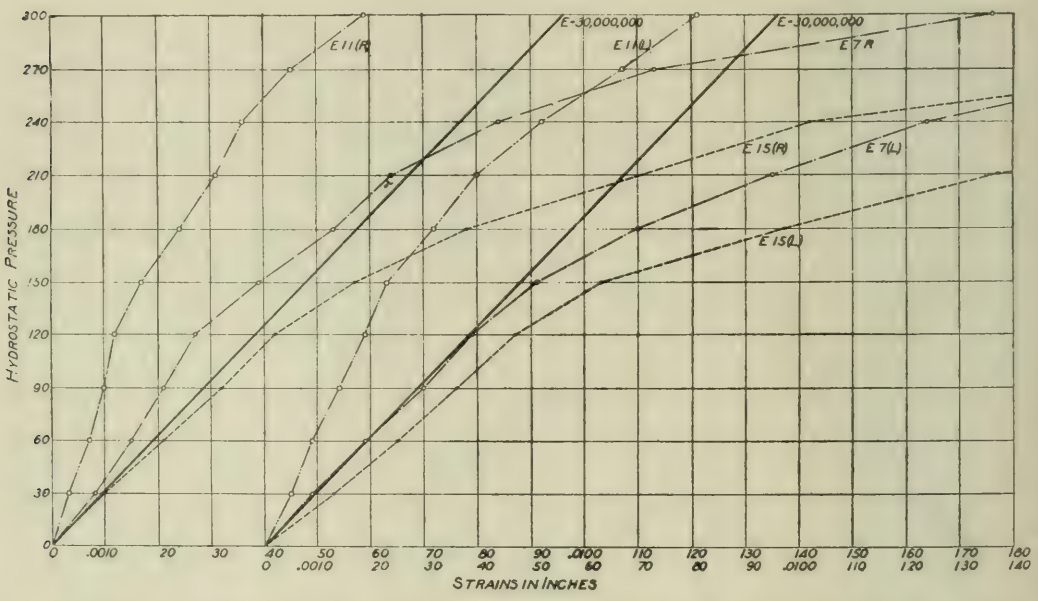


FIG. 25 CURVES OF TANGENTIAL EXTENSION, ACROSS LONGITUDINAL SEAMS, END COURSE *E*, EXTENSIONS NEAR REAR HEAD AND GIRTH SEAM AND MIDDLE OF LENGTH OF COURSE

46 The extension of course *A* at the edge over the front head was less than at the opposite edge. Gaged lengths *B*-52 and *B*-57 showed

the influence of the overlapping metal of the patch, as well as that of the girth seams. The extension at these two places was less than normal.

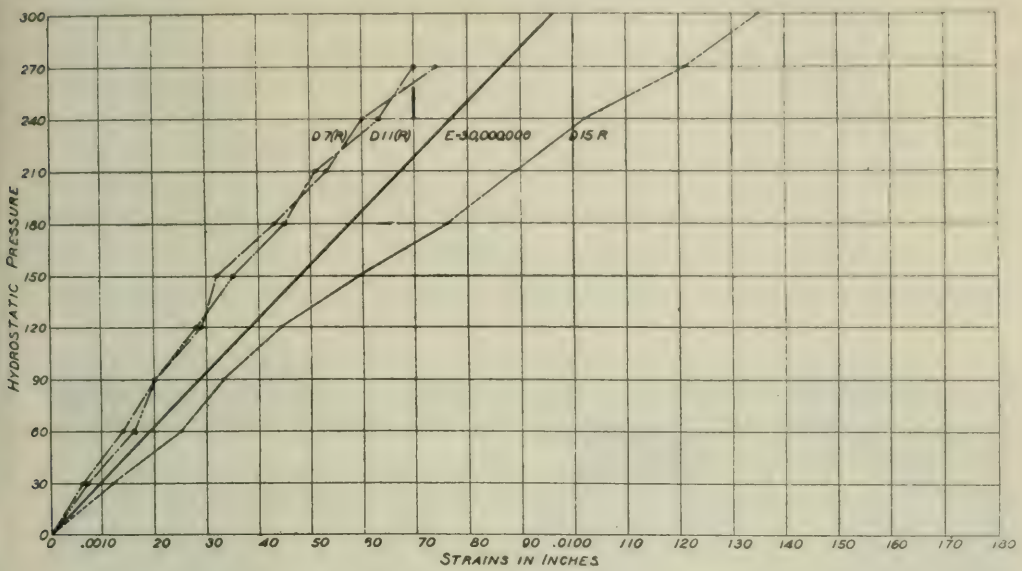


FIG. 26 CURVE OF TANGENTIAL RESILIENCE, ACROSS LONGITUDINAL SEAM, COURSE *D*, RIGHT SIDE OF BOILER, AT MIDDLE OF COURSE AND NEAR GIRTH SEAMS

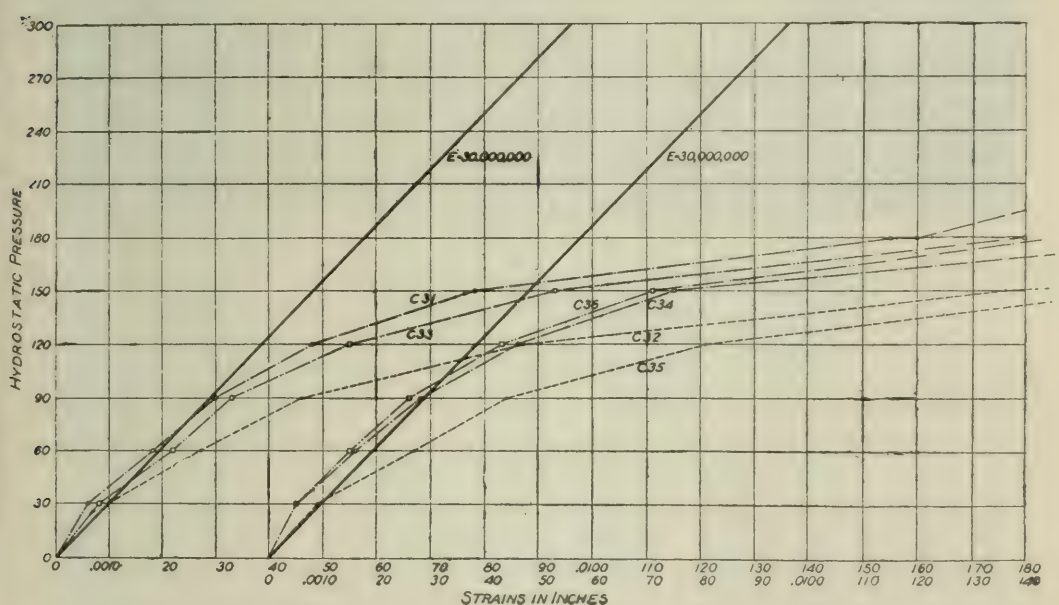


FIG. 27 CURVES OF TANGENTIAL EXTENSION, HAND-RIVETED SECTION, COURSE *C*, TOP OF BOILER

47 At the side of the manhole patch, *C*-58, there was found diminished rigidity in the shell. The weakness of this single-riveted patch was apparent in the measurements from the earliest pressures which

were applied to the boiler. Conditions about the safety-valve nozzle did not seem fully to compensate for this opening in the shell, as shown by the extensions on gaged lengths *E*-63 and *E*-64.

48 Referring next to the behavior of the shell at the seams, Fig. 22 shows two groups of curves of three lines each which represent the tangential extensions on gaged lengths established on course *C* taken across the longitudinal seam at the middle and at the edges of the course.

49 Curves *C*-7 and *C*-11, representing the extension at the edges of the course on the right side of the boiler, coincide in places and show but slight divergence where they depart most from the same line;

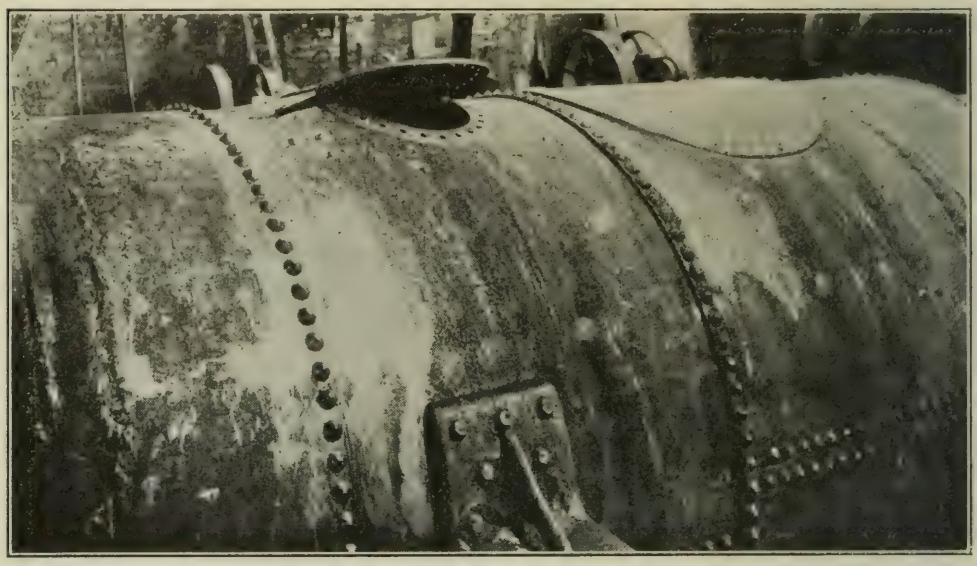


FIG. 28 MANHOLE PATCH AFTER RUPTURE. MAXIMUM PRESSURE ON SHELL 335 LB. PER SQ. IN.

that is, they indicate that uniform behavior was displayed at the opposite edges of this course. Each deflect rapidly under pressure above 150 lb. per sq. in., corresponding to a tensile stress of 14,400 lb. per sq. in. on the solid sheet.

50 At the middle of the length of the course, curve *C*-15 showed an increase in the rate of extension at the above-mentioned pressure, and for each succeeding pressure a greater extension than that witnessed at the edges. Necessarily, variations in the tangential extensions at different parts of the length of a seam would cause variations in the stresses of the solid metal of the shell in those localities.

51 In the case of a three-course boiler with one sheet to a course, as found in current construction, it would seem that a double-riveted

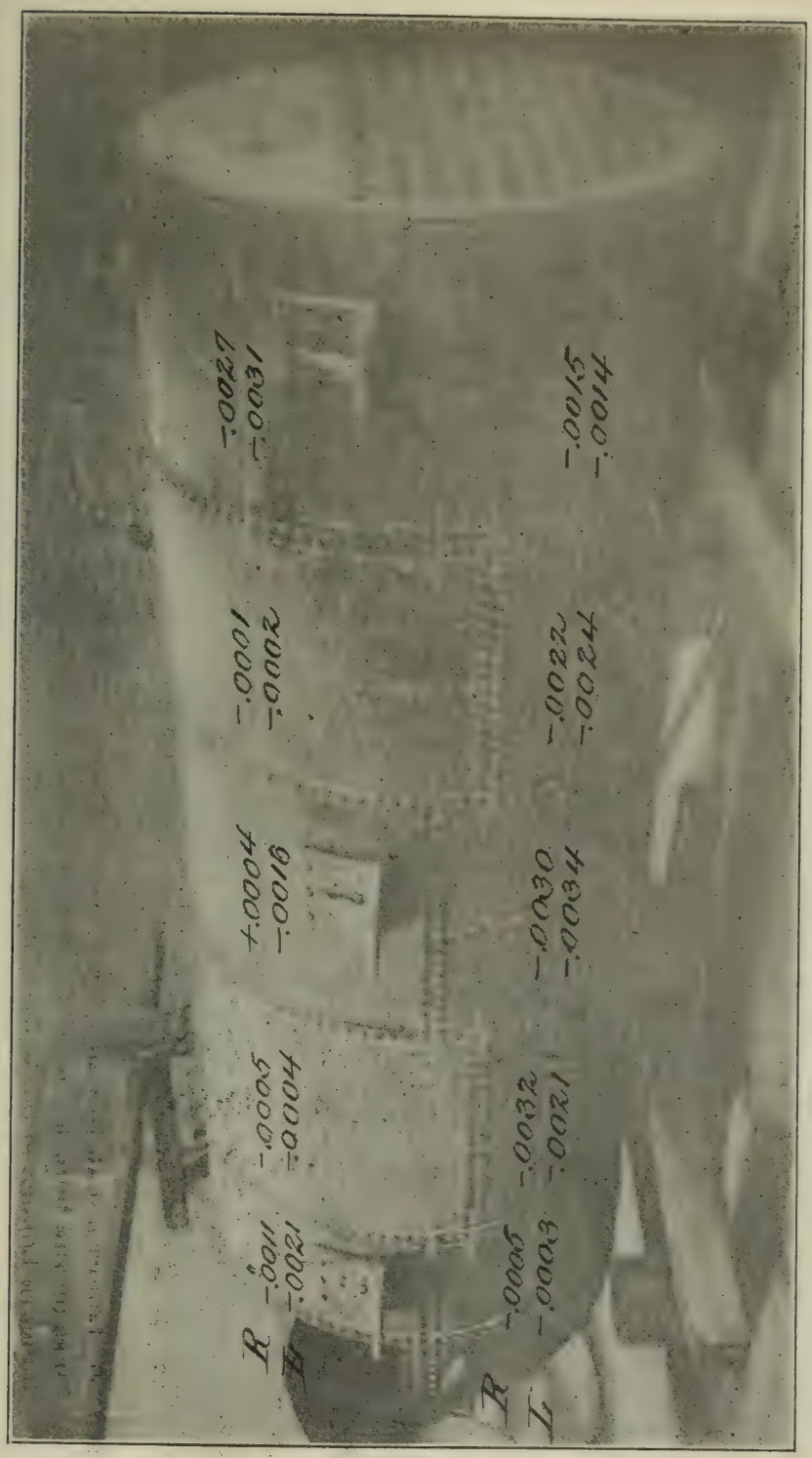


FIG. 29 LONGITUDINAL STRAINS ON SOLID SHEETS AT MIDDLE OF LENGTH OF COURSES, AT 270 LB. PRESSURE PER SQ. IN., RIGHT AND LEFT SIDES OF THE BOILER. MINUS SIGNS INDICATE LONGITUDINAL CONTRACTION; PLUS SIGNS LONGITUDINAL EXTENSION

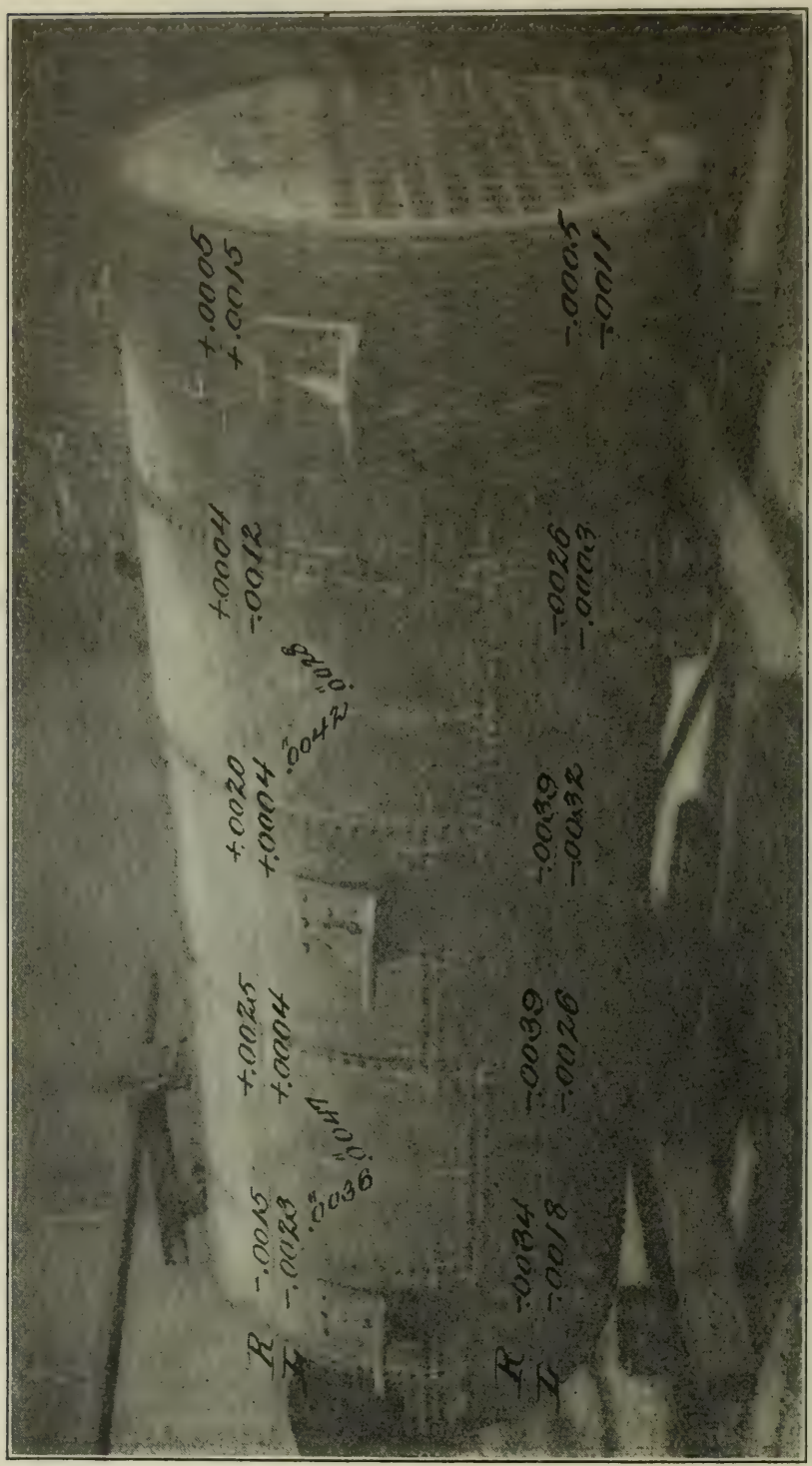


FIG. 30 LONGITUDINAL STRAINS, ACROSS GIRTH SEAMS, AT 270 LB. PRESSURE PER SQ. IN., RIGHT AND LEFT SIDES OF THE BOILER. ALSO STRAINS ON DIAGONAL GAGED LENGTHS OF COURSES *B* AND *D*. MINUS SIGNS INDICATE LONGITUDINAL CONTRACTION; PLUS SIGNS LONGITUDINAL EXTENSION

lap joint might occasion an excessive stress in the solid sheet abreast the end of the seam, under certain pressures.

52 In the present test the longitudinal seams, being only three rivet pitches apart, furnish a line from front to rear of the boiler across which the extensions are greater than those which are displayed by the solid sheets.

53 Fig. 23 shows results corresponding to those of Fig. 22 but pertaining to *D*, the next course of the boiler. The results are about the same on each, the maximum tangential extensions being displayed at the middle of the length of the seams.

54 Fig. 24 shows the extension of end course *A*, across the longitudinal seams on either side of the boiler and also the extension of

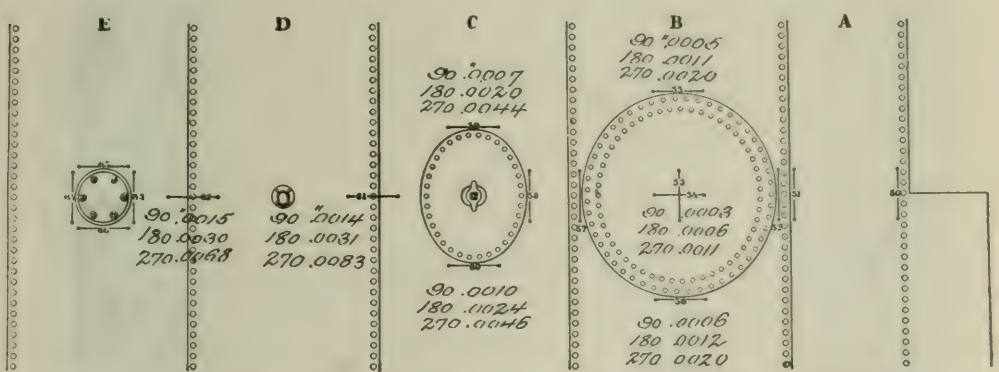


FIG. 31 LONGITUDINAL STRAINS, ON TOP OF BOILER IN VICINITY OF DOME AND MANHOLE PATCHES, ON DOME PATCH, AND ACROSS GIRTH SEAMS, AT 90, 180 AND 270 LB. PRESSURE

the solid sheet near the girth rivets of the front head. The divergent curves of this diagram indicate how differently in degree the metal is strained at the several gaged lengths of this narrow course. There seems, however, no lack of consistency in the behavior of the metal. The strains were relatively such as would be expected under the conditions present in this part of the boiler.

55 The strains in course *E*, at the rear end of the boiler are shown in Fig. 25, where the behavior of the shell was found to be similar to that at the front end. Notwithstanding the fact that the results appear consistent and the relations between the different parts of the boiler harmonious, attention is attracted by the variableness of the strains as they are found developed, according to the position of the measured lengths. The degree of variability witnessed in this type of boiler, which is certainly one of plain form, is such as to excite speculative interest in more complicated types.

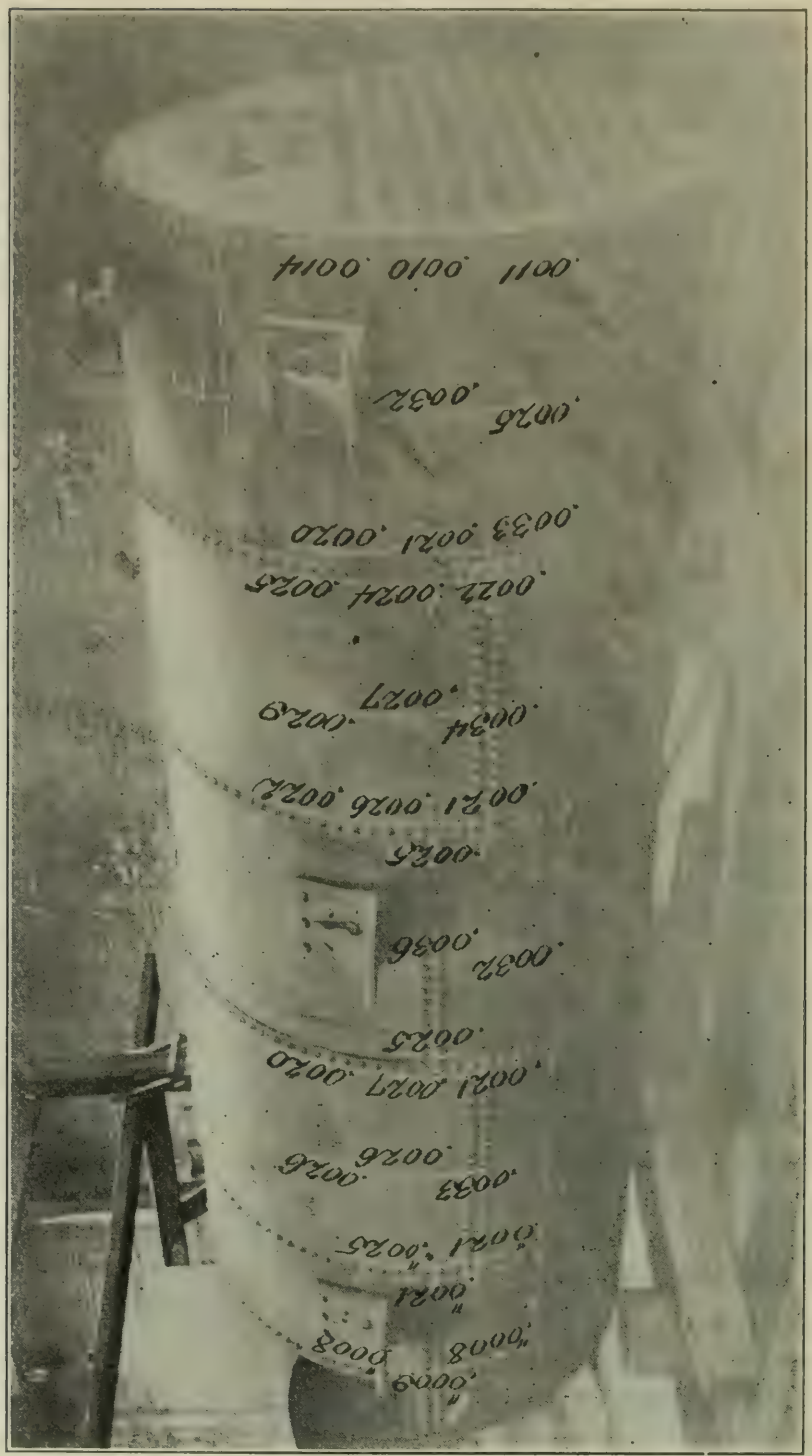


FIG. 32 TANGENTIAL EXTENSION AT 90 LB. PRESSURE. RIGHT SIDE OF BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0029 IN.

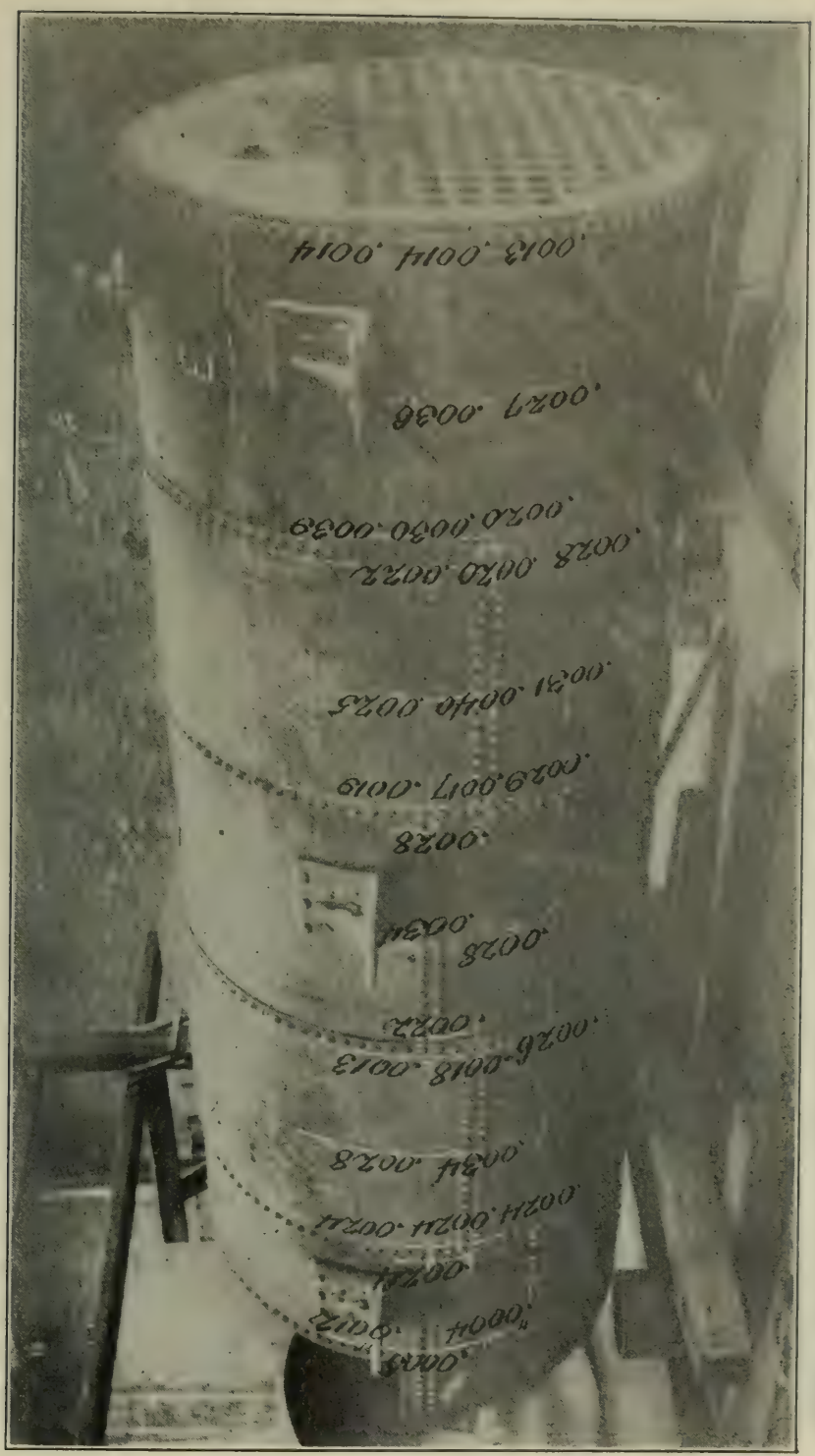


FIG. 33 TANGENTIAL EXTENSION AT 90 LB. PRESSURE. THE FIGURES GIVEN REPRESENT STRAINS DEVELOPED ON THE LEFT SIDE OF THE BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0029 IN.

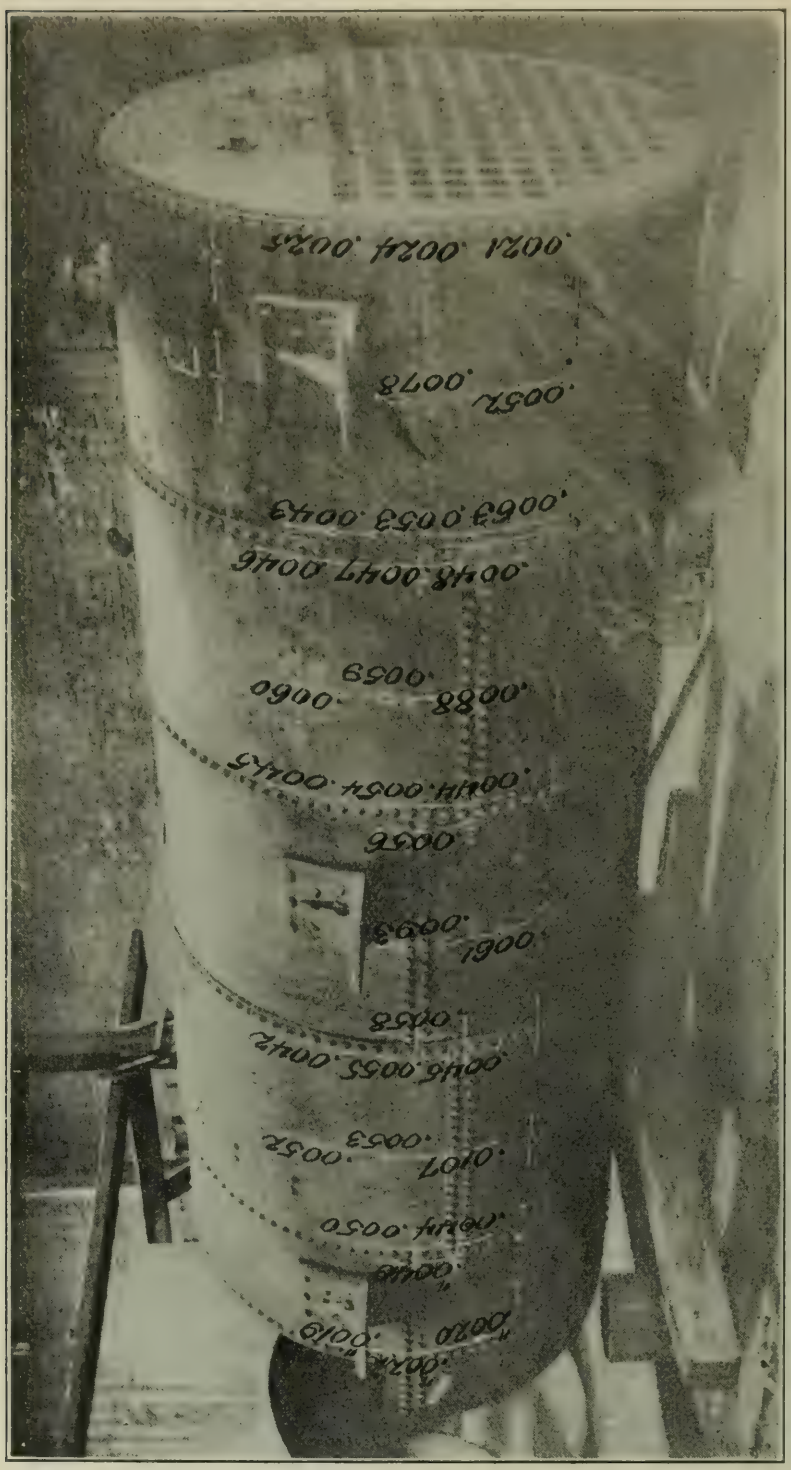


FIG. 34 TANGENTIAL EXTENSION AT 180 LB. PRESSURE. RIGHT SIDE OF BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0058 IN.

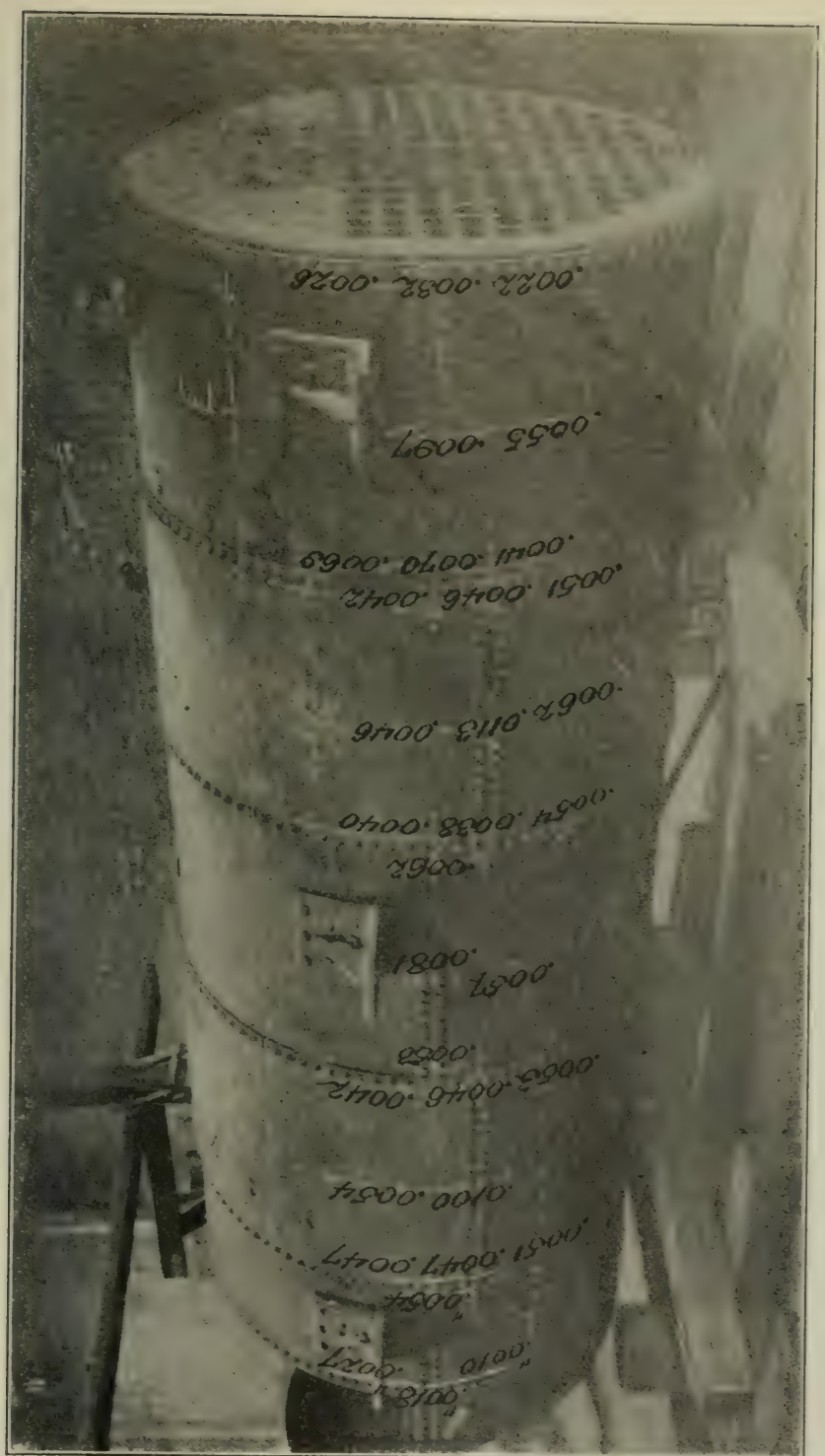


FIG. 35 TANGENTIAL EXTENSION, AT 180 LB. PRESSURE. THE FIGURES GIVEN REPRESENT STRAINS DEVELOPED ON THE LEFT SIDE OF THE BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0058 IN.

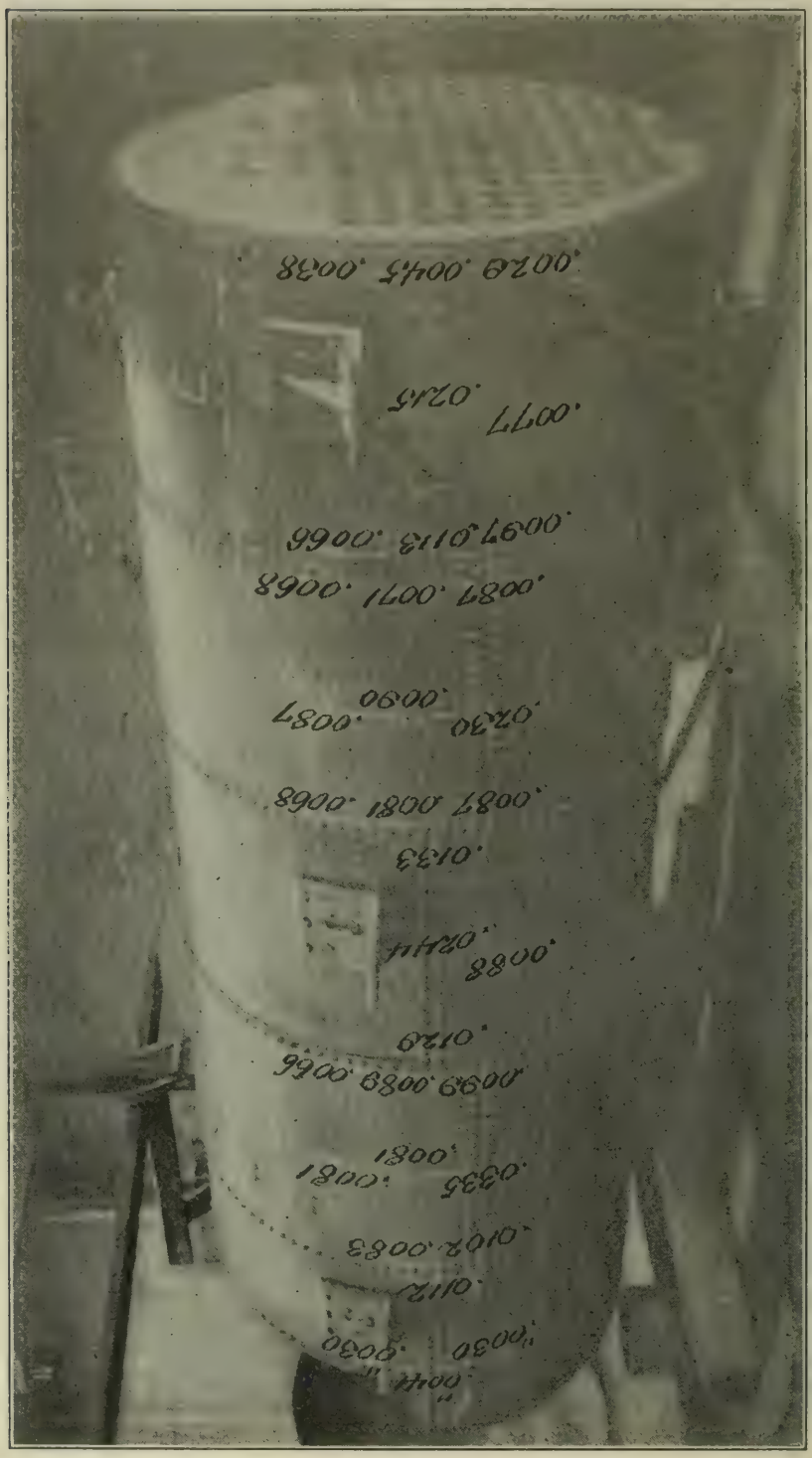


FIG. 36 TANGENTIAL EXTENSION AT 270 LB. PRESSURE. RIGHT SIDE OF BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0086 IN.

FIG. 37 TANGENTIAL EXTENSION AT 270 LB. PRESSURE. THE FIGURES GIVEN REPRESENT STRAINS DEVELOPED ON THE LEFT SIDE OF THE BOILER. NORMAL, COMPUTED EXTENSION AT THIS PRESSURE, 0.0086 IN.

56 The results of the diagrams, Figs. 18–27, refer to the total extensions of the gaged lengths, that is, they include the elastic extensions and the permanent sets when sets have occurred. The curves in Fig. 26 were plotted for the purpose of showing the elastic

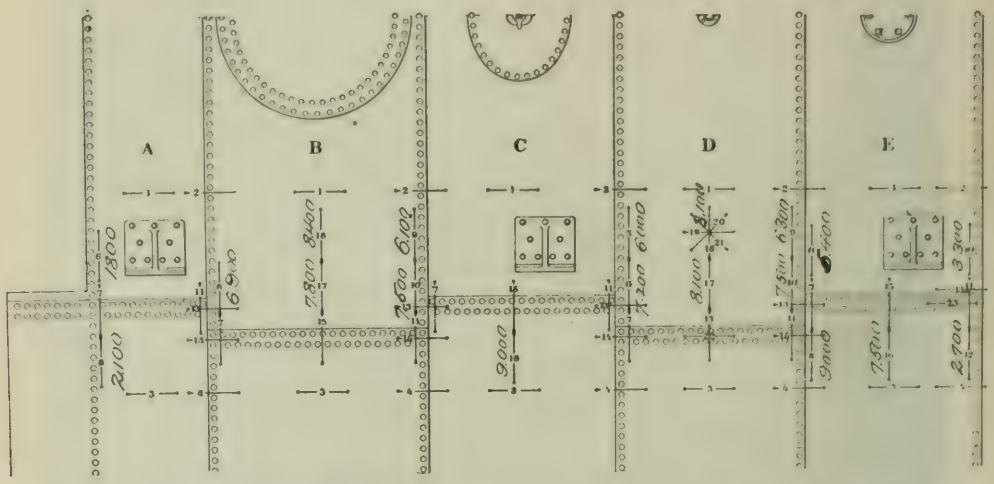


FIG. 38 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 90 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. RIGHT SIDE OF BOILER. NORMAL COMPUTED STRESS, 8640 LB. PER SQ. IN.

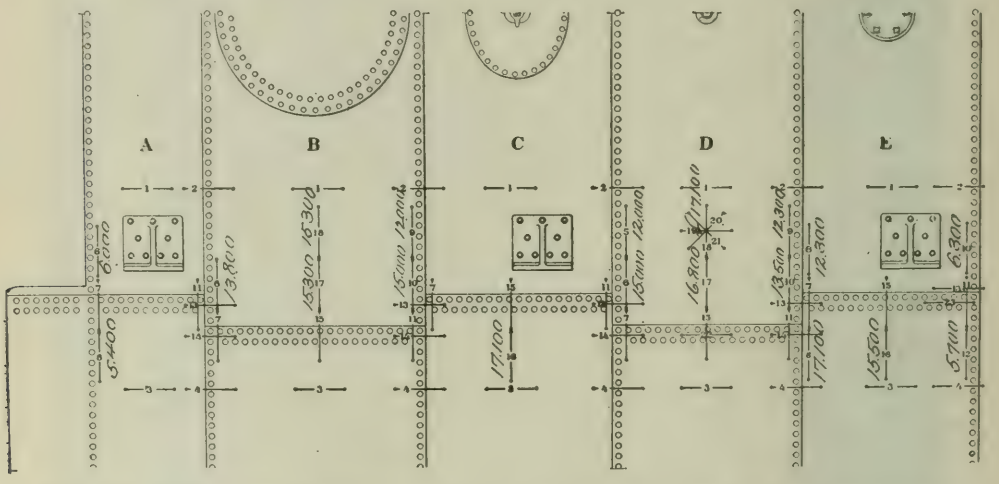


FIG. 39 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 180 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. RIGHT SIDE OF BOILER. NORMAL COMPUTED STRESS, 17,280 LB. PER SQ. IN.

extensions only, or what is equivalent to the same, the resilience of the shell taken across the longitudinal seam of course *D*, right side. The greater resilience of gaged length *D*-15, which was located across the longitudinal seam at the middle of its length, over the amount called for by the modulus of elasticity curve will be noted. This might

be taken to indicate an intensity of stress above the normal in the shell in that vicinity, or it may mean that bending and shearing stresses at the seam in addition to tensile stresses on the sheets modified the results.

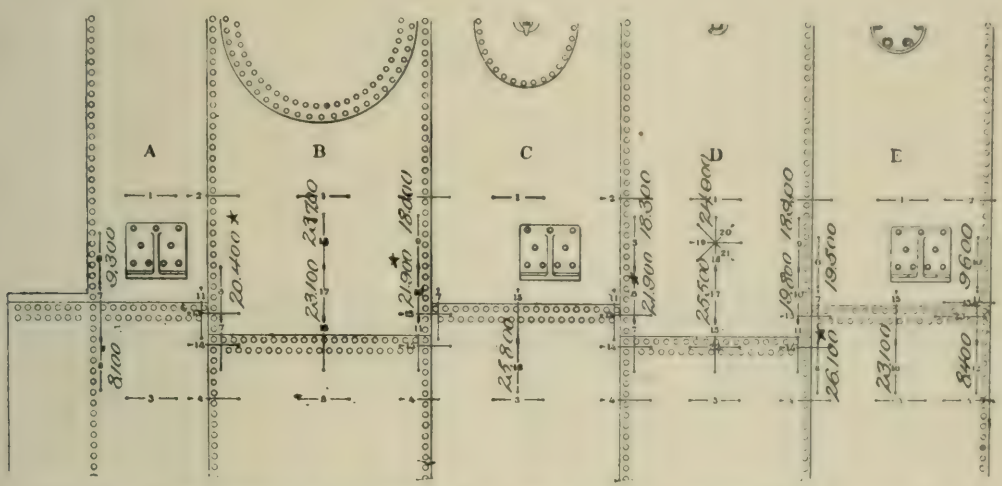


FIG. 40 TANGENTIAL STRESSES, LB. PER SQ. IN. AT 270 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. RIGHT SIDE OF BOILER. NORMAL COMPUTED STRESS, 25,920 LB. PER SQ. IN. GAGED LENGTHS ON WHICH DECIDED PERMANENT SETS OCCURRED, INDICATED BY A STAR

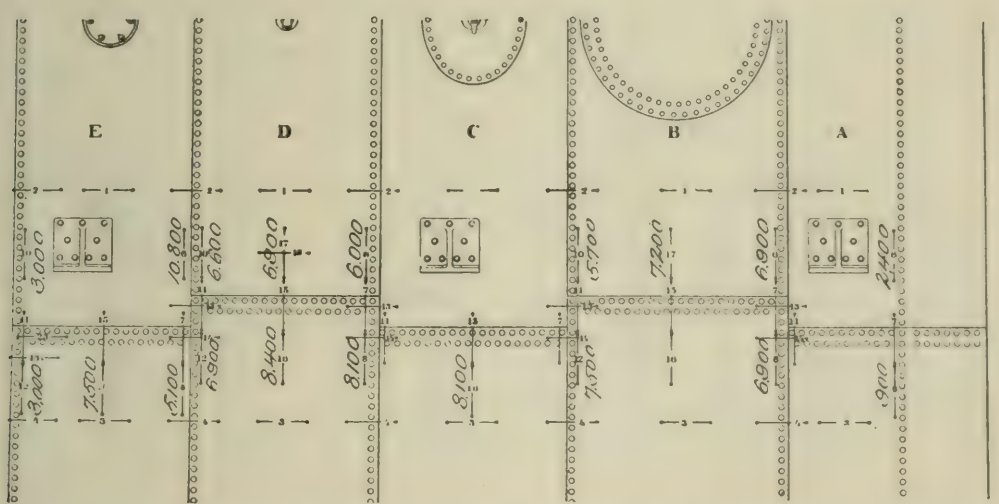


FIG. 41 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 90 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. LEFT SIDE OF BOILER. NORMAL COMPUTED STRESS, 8640 LB. PER SQ. IN.

57 The interior pressure on the boiler was increased from 300 lb., the highest indicated on the diagrams, to 335 lb., under which latter pressure rupture of the manhole patch occurred. Three of the rivets were sheared by the tangential stress of the shell, followed, apparently,

by the fracture of other rivets by tension on the stems, which pulled off the heads and finally tore the shell longitudinally along its upper element, starting this fracture at a rivet hole of the manhole opening. Fig. 28 shows the appearance of this fracture.

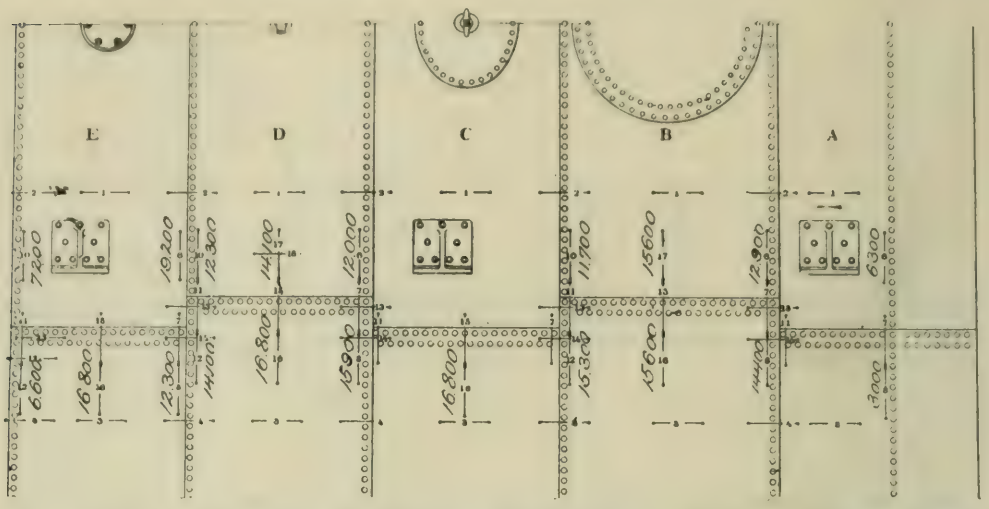


FIG. 42 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 180 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. LEFT SIDE OF BOILER. NORMAL COMPUTED STRESS, 17,280 LB. PER SQ. IN.

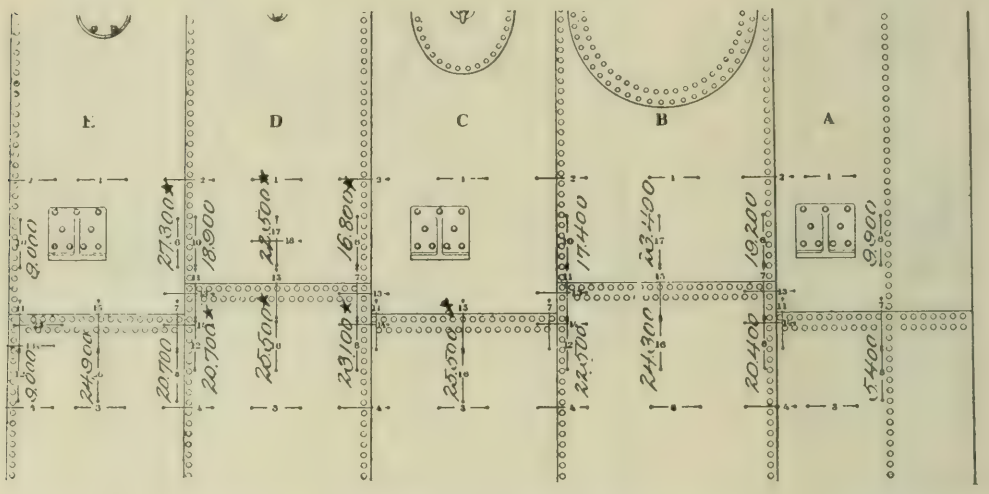


FIG. 43 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 270 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. LEFT SIDE OF BOILER. NORMAL, COMPUTED, STRESS, 25,920 LB. PER SQ. IN. GAGED LENGTHS ON WHICH DECIDED PERMANENT SETS OCCURRED, INDICATED BY A STAR

58 The shell was repaired by cutting out a portion of course *C*, across the top of the boiler and putting in a section the full length of the course and about 3 ft. wide, measured on the arc. This new

section was double-riveted to the shell at its longitudinal seams. The rivets were $\frac{13}{16}$ in. in diameter and had a pitch of 2.87 in. The rows were 1.53 in. apart, with rivets staggered, which, were, of course, hand-driven. The points of the rivets were hammered down to conical shape, low in height and with thin edges. In this respect they were less substantial than the points of the original machine-driven rivets of the seams.

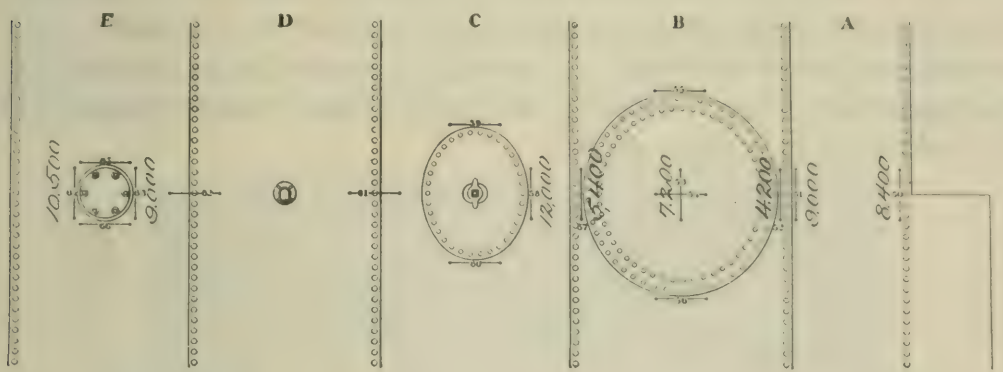


FIG. 44 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 90 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. TOP OF BOILER. NORMAL COMPUTED STRESS, 8640 LB. PER SQ. IN.

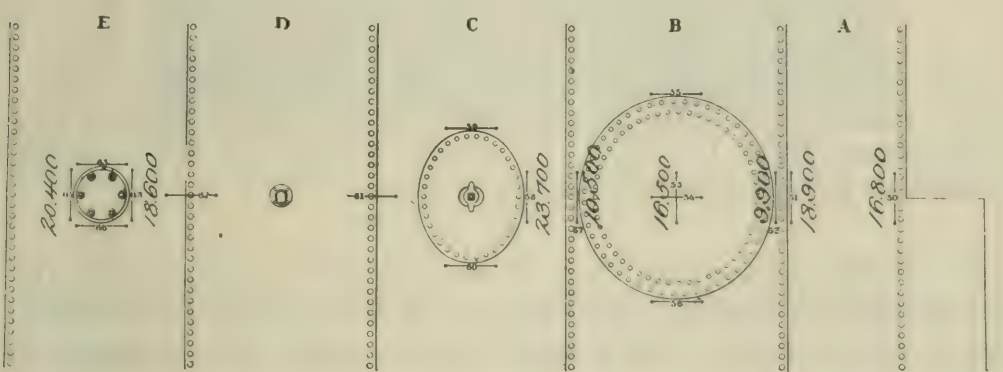


FIG. 45 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 180 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. TOP OF BOILER. NORMAL COMPUTED STRESS, 17,280 LB. PER SQ. IN.

59 The hand-driven rivets would not be expected to hold the calking as well as the machine-driven rivets by reason of the difference in their points, and the test showed that such weakness was the case.

60 The diagram, Fig. 27, shows the behavior of the seams of this new section of course *C*. The flatness of the curves indicate how early these new seams began to display rapid extension, and with so decided a movement the calking was soon disturbed and copious leaks started.

At the time of presenting these notes no higher pressure has been reached than the rupturing one of 335 lb. previously mentioned.

61 The strain measurements thus far described were those which were observed on tangential gaged lengths. In addition to this as indicated in the diagrams, Figs. 15, 16 and 17, there were longitudinal gaged lengths laid off on the shell and measured.

62 In a plain cylindrical shell the tangential extension of the metal would necessarily be attended with a definite amount of longitudinal contraction, eliminating the effect of pressures on the head. The conditions, however, which are present in steam boiler construction will generally prevent realizing the longitudinal strains which would be looked for in a plain sheet.

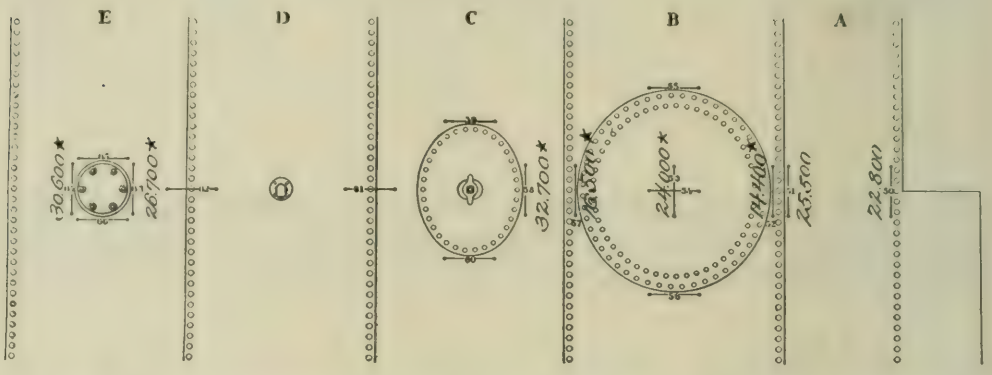


FIG. 46 TANGENTIAL STRESSES, LB. PER SQ. IN., AT 270 LB. BOILER PRESSURE. RESULTS BASED ON RESILIENCES. TOP OF BOILER. NORMAL COMPUTED STRESS, 25,920 LB. PER SQ. IN. GAGED LENGTHS ON WHICH DECIDED PERMANENT SETS OCCURRED, INDICATED BY A STAR

63 In the present test there were parts of the boiler nearly free from longitudinal strains, while there were other places in which the strains were reversed, and longitudinal extension shown instead of longitudinal contraction.

64 In order to determine whether the action immediately at the girth seams was represented by the 10-in. gaged lengths which spanned them symmetrically, other gaged lengths were established on the shell, not indicated on the diagrams herewith presented. These were in pairs, one being wholly on the solid sheet, the other just stepping on to the adjacent course. The observations on these gaged lengths lead to the same results, however, as found on those which symmetrically spanned the seam.

65 The results of these observations showed that along the lower quarter of the boiler the longitudinal strains were contractions;

while along the upper quarter they were in part contractions, and in part extensions. The strains observed at 270 lb. pressure are entered on two lightly printed photographs, Figs. 29 and 30. In Fig. 29 are shown the strains which were measured on gaged lengths 1 and 3, taken on the solid sheets at the middle of the lengths of the courses. In Fig. 30 are shown the strains which were measured on gaged lengths 2 and 4, taken across the girth seams. Minus signs before the figures indicate contractions while plus signs indicate extensions.

66 It will be observed that the lower part of the shell contracted longitudinally, notwithstanding the fact that the tubes were extended by reason of the exterior pressures to which they were subjected.

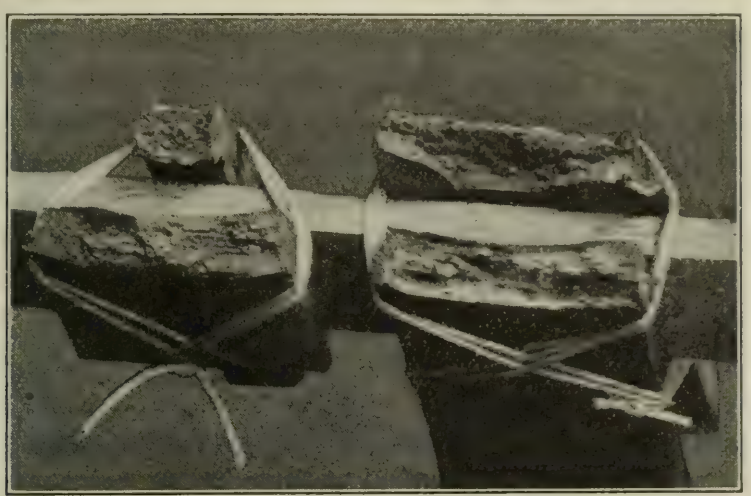


FIG. 47 LAMELLAR APPEARANCE OF METAL, SHEET *C*, FRACTURED BY BENDING WHILE AT A BLUE HEAT

67 This behavior calls for bending at the flanges of the heads to compensate for the difference in direction of these movements. The six through braces would relieve the shell of a portion of the longitudinal tension coming from the heads in the upper half of the boiler.

68 Longitudinal gaged lengths on the upper part of the shell showed diminished contractions over those observed on the lower portion, or displayed strains of extension. On the very top of the boiler the strains were extensions of a pronounced order.

69 It was found on diagonal gaged lengths, laid off on courses *B* and *D*, upper quarter of the boiler, that greater extensions were displayed on the converging diagonals over those of diverging directions. The converging diagonals, at 270 lb. pressure, extended 0.0047 in. and 0.0042 in. respectively, against 0.0036 in. and 0.0028 in. displayed on the diverging gaged lengths.

70 The location of the diagonal gaged lengths on course *D* are shown in Fig. 14, similar lengths having been laid off on course *B*. These results are entered in Fig. 30, in addition to the longitudinal strains. The longitudinal strains, all being those of extension, observed on the top of the boiler, have been entered on the diagram, Fig. 31. The strains for each pressure, 90, 180, and 270 lb. respectively, are given. The range and variability of these measurements are seen to be very pronounced.

71 A series of lightly printed photographs, Figs. 32 to 37 inclusive, are presented, on which are entered the measured tangential strains observed at pressures of 90, 180 and 270 lb. respectively.

72 The strains observed on both the right and the left sides are entered, however, on prints representing the right side of the boiler. It will be understood that the longitudinal seams of the left side were the reverse of those on the right side as regards their respective heights. That is, on the left side the seams of course *B* and *D*, were three rivets above instead of three rivets below the others, as shown in illustrations of the right side. The tangential strains called for by computation based on a 30,000,000 modulus of elasticity are stated for each pressure in the captions of the figures.

73 The range in measured strains above and below the computed amount will be noted upon inspection of Figs. 32–37. The strains were least in amount at the heads, and generally greater rigidity was displayed at the intermediate girth seams than in the solid sheets, under the earlier pressures. At the middle of the length of the longitudinal seams the maximum extensions were developed, witnessed in this examination of the behavior of the shell.

74 The results thus far presented, with the exception of those on Fig. 26, have included both the elastic strains and the permanent sets of the different measured lengths.

75 The permanent sets have been subtracted from the extensions and the stresses corresponding to these resiliences computed, and the results entered on another series of diagrams, Figs. 38 to 46 inclusive. These results show the tangential stresses, in pounds per square inch, found in different parts of the solid sheets of the shell when the boiler was subjected to pressures of 90, 180 and 270 lb., respectively.

76 In looking over those results the usual influence of longitudinal seams, such as were used in this boiler, in intensifying the tangential stresses in the adjacent solid sheets, may be pointed out. The excessive stress at the side of the single-riveted manhole patch is

clearly shown in the results. The high stresses at the sides of the safety-valve nozzle under the maximum pressure will also be noted.

77 While the results are consistent, nevertheless as an engineering structure the distribution of stresses exhibits a range far beyond that which is expected in other classes of constructive work. The type of boiler being one of the simplest, the extension of this method of test to other types would seem desirable. Such tests might assist in establishing satisfactory rules for steam boiler construction and might reasonably be expected to aid in the framing of regulations governing allowable pressures.

78 Additional tests were carried out, in which the effect of changes in the manner of supporting the boiler was inquired into. It was supported on the four end lugs in one test, and again in another test most of the weight was carried by the middle lugs. In each case there was a modification in the measured strains, although not in a marked degree. At the end of the test the several courses were seen to have been visibly extended in diameter between the girth seams.

79 The chemical composition of the steel in course *C* was as follows:

Carbon	0.22
Manganese	0.43
Silicon	0.046
Sulphur	0.028
Phosphorous	0.043

It is recalled that this particular brand of steel at the time of its manufacture was not infrequently found to possess a decidedly laminated structure. The laminations were not large, nor likely to cause blisters in the boiler, but they were in places quite numerous.

80 The metal from course *C*, the only sheet yet examined, was found to have a laminated structure. Fig. 47 shows the appearance of some fractured strips from this course, which were bent while at a blue heat, in order to develop the lamination of the plate in a pronounced manner. The metal drifts well, a $\frac{3}{4}$ in. diameter punched hole having been drifted cold to $1\frac{1}{4}$ in. diameter without rupture.

81 The services of Mr. P. W. Brunner, and J. W. Herrity are acknowledged, whose skill as manipulators is shown by the internal evidence of reliability which these measurements, taken by them, furnish.

DISCUSSION

FRANCIS B. ALLEN. The paper contains some very interesting data which are more complete than any we have ever had so far as I know, in tests of boilers of this description. In my own experience, it is a very difficult problem to convince a manufacturer when a battery of boilers has given good service and, superficially at least, is without patches or other evidence or deterioration, that it is necessary to reduce pressure in order to maintain so far as possible the full factor of safety. I have in mind a set of boilers which was in operation for 27 years, in which the original pressure allowed was 85 lb. By successive cuts as the boilers increased in age, this pressure was reduced to 60 lb. and upon inspection a recommendation was made that the pressure be reduced to 50 lb., or preferably that newer and stronger boilers be submitted. This meant an expense of thousands of dollars and the manufacturers naturally wished to use the boilers a little longer and so make a greater return upon the investment.

In the belief, however, that it would not be advantageous to take the risk involved, they were taken out; and the test which the paper describes established the fact that while there was an ample factor of safety according to the requirements of law and custom, the recommendation to remove the boilers was fully justified because they had approached the danger line as closely as should be allowed. The fact that these boilers described in the paper were exceptionally good and had been carried by one organization and insured by another from the time they were put into commission until they were taken out, makes the record and data unusually valuable.

THE AUTHOR. The two boilers under consideration present features of interest because of the long term of their service and their excellent condition when the tests were made. Originally, these boilers were run under 90 lb. steam pressure which was eventually reduced to 60 lb., the pressure carried at the time they were taken out of service.

An inspection of the interior of the first boiler showed little or no scale on the sheets, excepting a limited amount of loose material on the bottom of the boiler. The upper row and side vertical rows of the tubes were free from deposit, while the interior ones had a deposit on the rear quarter or half and at their immediate front ends. The tubes, however, were free from pitting and general corrosion.

The condition of the boilers was such that the results of the strain measurements may be taken as representative of the behavior of boilers of their type in respect to the distribution of stresses in different parts of the shell when under pressure.

In the test of the first boiler, such copious leakage at the single riveted longitudinal seam of the dome occurred at 266 lb. pressure that its removal was necessitated. At that time, the metal at the base of the dome was overstrained and lines of scale were detached, showing that the elastic limit of the metal had been passed. The manhole frame was the next to yield, fracturing at 270 lb. pressure. Several of the stays at the front head at 295 lb. pressure ruptured. The test was then discontinued and the boiler dismantled. The metal from the shell and some of the tubes was reserved for subsequent examination and test.

Prior to testing the second boiler it was divested of its dome and manhole frame and the heads were strengthened by means of six through-braces. With this boiler a rupturing pressure was reached at 335 lb., the rupture occurring at the single-riveted patch covering the manhole opening. A section of this course was cut out and the shell repaired with a new piece of plate, which was double-riveted. Hand-holes necessary for riveting were made in the new plate. No higher pressure has yet been reached than the earlier one of 335 lb., leakage overcoming the pumps and taking place at the longitudinal seams and particularly at the new hand-riveted seams.[1]

Concerning the distribution of the stresses in the shell, at few of the measured lengths was the tangential extension of the shell found to correspond closely to the computed extension, although there were places where there was full agreement. Generally, the tangential extensions in the vicinity of the girth seams were less than the computed values. Naturally the edges of the end courses over the heads displayed only a limited amount of extension.

Across the longitudinal seams, taken at the middle of the lengths of

[1]A pressure of 370 lb. has since been reached. The sheets show further deformation but have not yet ruptured.

the several courses, the extensions were greater than called for by computation for the solid sheets. This increased yielding at the longitudinal seams became more pronounced as higher pressures were successively applied, the extensions in the vicinity of the girth seams, however, remaining less than the normal amount.

A place characterized by unusual rigidity was found on the top of the boiler in the immediate vicinity of the double-riveted manhole patch and the girth seam, while in the next course at the side of the manhole patch unusual extension was displayed.

The relative rate of extension observed under early pressures was maintained without material change under the subsequent higher pressures.

The longitudinal strains were found to be contractions on the lower part of the shell, in part contractions and in part extensions on the upper part of the shell, while on the top of the boiler they were all strains of extension.

The method of measured strains illustrated in these results is a convenient one for extending the scope of research work beyond the usual limits of laboratory tests as conducted in testing machines. Knowledge pertaining to the distribution of stresses, important in structures of all kinds, may be ascertained in this manner. The constructive merits of different types of boilers admit of being ascertained by the method of measured strains.

It would seem feasible, judging from the indications of the present tests, to determine the merits of different types by means of observations confined within the limits of the usual hydrostatic test pressures, as applied to new boilers. That is, an adequate opportunity is presented in the hydrostatic test to acquire this information and in no wise impair the strength of the shell or other parts by reason of having made such strain measurements.

The obvious value of such information will not require further mention. It is believed that herein is presented a method of test which will lend material aid in furnishing a rational basis on which to establish rules regarding allowable steam pressures.

With the present knowledge available on the effects of repeated stresses in destroying the integrity of structures, it is apparent that information pertaining to the behavior of structures under working loads is desirable. Most failures emphasize the need of more complete data on working conditions rather than any deficiency of knowledge on the ultimate strength of the component parts taken individually.

No. 1330

HERRINGBONE GEARS

WITH SPECIAL REFERENCE TO THE WUEST SYSTEM

By Percy C. Day,[1] Milwaukee, Wis.

Non-Member

That the helical principle in toothed gearing is ideal from a theoretical point of view is well known. From a practical standpoint herringbone gears have been less satisfactory than straight-cut spur gears because, until recently, no method was devised for producing them with the requisite speed and accuracy. Within the last six years a method has been found and developed in England to a high degree of perfection. Herringbone gears made by this method are called Wuest gears, after the name of the inventor, and can be produced with even greater accuracy than cut gears of the spur type (Figs. 1 and 2).

2 The distinction between these gears and those of the ordinary herringbone type is that the teeth of the former, instead of joining at a common apex at the center of the face, are stepped half the pitch apart and do not meet at all. This arrangement of the teeth does not affect the action of the gears, but it facilitates their commercial production and admits the use of precision methods in their manufacture.

3 The utilization of power constantly calls for means to transmit rotary motion from one axis to another and to transform the speed of rotation. While there are many ways in which such transmissions and transformations may be produced, the merits of all of them must be judged from the standards of (*a*) reliability and freedom from wear and tear; (*b*) economy of outlay; (*c*) mechanical efficiency; (*d*) compactness; (*e*) evenness of transmission, absence of shock, jar or vibration; (*f*) absence of noise. The order of merit may change with different applications, but the same standards obtain for all

[1] Engineer with The Falk Co.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

transmissions. The various methods employed are so well known that they need no discussion here. Let it suffice to say that the spur gear, which satisfies only the first four conditions, is used to such an extent that all other appliances are relatively unimportant.

ACTION OF SPUR GEARING

4 The aim of all designers of gearing is to transmit rotary motion

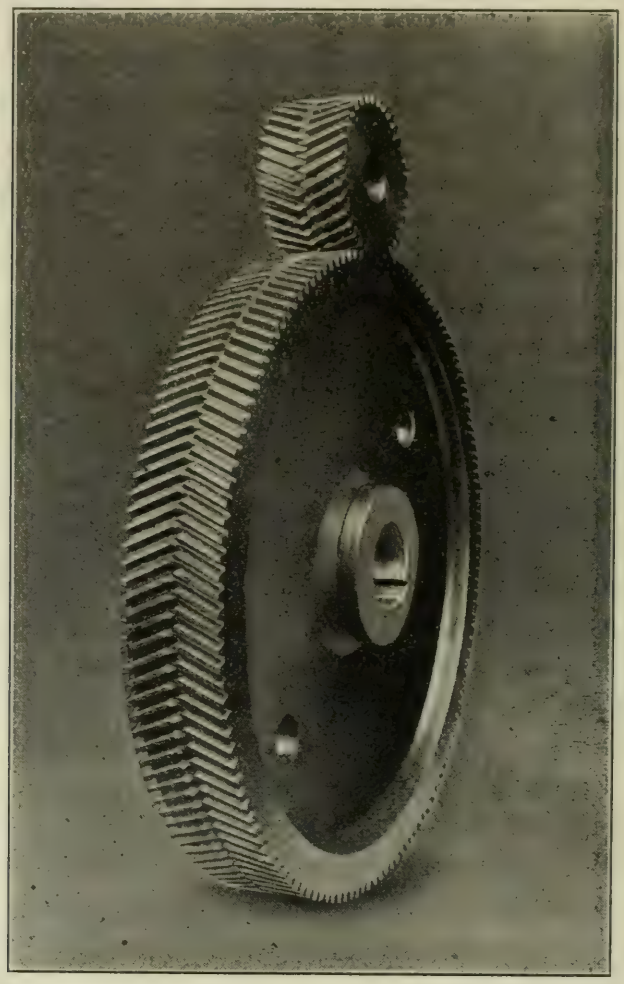


FIG. 1 TYPICAL HERRINGBONE GEAR AND PINION

from one axis to another in a perfectly even manner without variation of angular velocity. Let us consider the action of a straight spur pinion driving a gear. There are three distinct phases of engagement:

a First phase: The root of the pinion tooth engages the point of the gear tooth.

b Second phase: The teeth are engaged near the pitch line.

c Third phase: The point of the pinion tooth engages the root of the gear tooth.

5 Let us assume that the teeth are accurately cut to involute form, so that if the pinion moves with even angular velocity it will produce corresponding evenness of motion in the gear. Also that the pinion has sufficient teeth to allow the engagement of successive teeth to overlap.

6 At the beginning of the first phase, while the load is carried

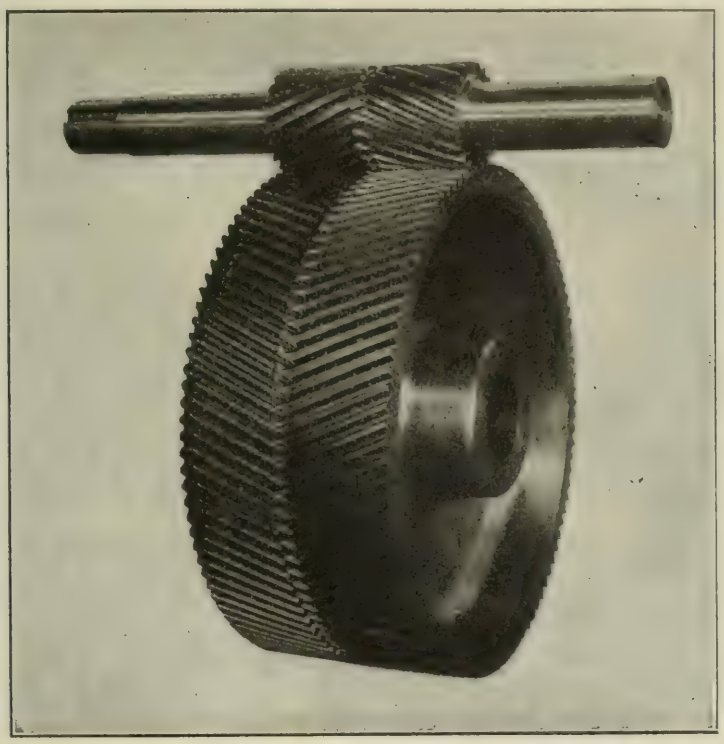


FIG. 2 GEAR AND PINION FOR 200-H.P. TURBINE, TO RUN AT 5000 FT. PER MIN.

near the point of the gear tooth, that tooth is subjected to a maximum bending stress along its whole length. During the first phase, the portion of the pinion tooth near the root is sensibly sliding over the outer portion of the gear tooth; that is to say, two metallic surfaces of small area are sliding under heavy compression.

7 The action during the second phase more nearly approaches ideal conditions. The teeth are engaged near their respective pitch lines and very little sliding takes place.

8 During the third and final phase, the pinion tooth is subjected to a maximum bending stress, while the tooth surfaces again slide

over each other, this time with the outer portion of the pinion tooth engaging the gear tooth near its root.

9 The point to be noted is that while those portions of the mating teeth which are near the pitch lines transmit the load with rolling contact, those which are more remote have to transmit the same load with sliding contact. The inevitable result is that the points and roots of all the teeth tend to wear away more rapidly than the portions near the pitch lines, so that the involute tooth curves, necessary to the preservation even of angular velocity, lose their form and the motion becomes uneven.

10 It may be suggested that the sliding action can be obviated by shortening the teeth so that they engage only the phase of rolling contact. This has been tried with a certain measure of success in the stub-toothed gear, but it cannot be carried far enough without curtailing the arc of contact so that continuity of engagement is lost, thereby introducing more serious trouble than that which it is desired to avoid.

11 Distortions of gear teeth from involute form, whether due to inaccurate cutting or subsequent wear, give rise to all kinds of trouble. The average angular velocity may be uniform, and yet the passage of each pinion tooth through its brief engagement with the mating gear may be accompanied by successive retardation and acceleration which, though small in itself, takes place in such a short interval of time that it may cause interacting stresses many times greater than the average working load on the teeth. These internal stresses are very difficult to deal with, because they are indeterminate. They cause noise, vibration, crystallization and fracture.

ACTION OF HERRINGBONE GEARS

12 Herringbone gears completely overcome all these difficulties, but only when they are accurately cut.

13 The writer will first assume the accuracy and describe the action, afterwards he will endeavor to show in the system under discussion the special features which insure the production of accurate herringbone gears on a commercial scale.

14 If we take two exactly similar pinions with straight teeth and place them side by side on one shaft, with the teeth of one pinion set opposite the spaces of the other, then we have what is known as a stepped-tooth pinion. If this pinion is meshed with a composite gear made up in a similar manner, the action is modified so that there are always two phases of engagement taking place simultaneously.

Such gears are commonly used for rolling-mill work, because they stand up to heavy shocks better than the plain type.

15 Still better action can be secured by assembling a number of narrow pinions with the last of the series one pitch in advance of the first and the others advanced by equal angular increments. As a practical proposition, however, gears made on these lines would be costly and difficult to produce.

16 The helical gear is the logical outcome of the stepped gear carried to its limit, and built up from infinitely thin laminations. Since the steps have merged into a helix, there must be a normal component of the tangential pressure on the teeth, producing end thrust on the shafts. To obviate end thrust, the helical teeth are made right-hand on one side of the face and left-hand on the other. Such gears, with double helical teeth, are known as herringbone gears.

17 The fundamental principle of the action of herringbone teeth lies in the circumstance that *all phases of engagement take place simultaneously.* This holds good for every position of pinion and gear, provided only that the relationship between pitch, face width, and spiral angle is such as will insure a complete overlap of engagement.

18 Since all phases of engagement occur together, it follows that the load is partly carried by tooth surfaces in sliding contact and partly by surfaces in rolling contact. The result is curious and interesting.

19 Those portions of the teeth farthest from the pitch line, which engage with sliding action, tend to wear away more rapidly than the portions nearest the pitch line. But the pitch line portion is always carrying part of the load, and the effect of wear on the ends of the teeth merely tends to throw more load on the center portions; in other words *there is a tendency to concentrate the load near the pitch lines.* The ends of the teeth, instead of wearing away to an ever-increasing extent from their original involute form, are relieved of some of the load from the moment that wear commences to take place. As soon as the load on these ends has been partially relieved and transferred to the middle portion, the wear becomes equalized all over the teeth and they do not tend to distort further from their original shape.

20 It is quite clear that an immeasurable amount of wear on the tooth ends will be sufficient to relieve them of all the load, so that the distortion from original form will be practically nothing. The minute extra wear that does take place at the ends is only the amount necessary to transfer a proportion of the load near the

pitch lines sufficient to equalize the wear all over the surface of the teeth, the portions in sliding contact carrying less than those in rolling contact.

21 Thus the teeth keep their involute form, and motion is transmitted from pinion to gear in a perfectly even manner, without jar, shock, or vibration. Although herringbone teeth may not be intrinsically stronger than straight teeth, the elimination of all shock and indeterminate internal stresses renders them capable of dealing with far heavier transmitted loads. The concentration of the major portion of the load on the parts of the teeth in rolling contact eliminates friction to a marked extent.

22 Since all phases of engagement occur simultaneously, the transference of the load from one pinion tooth to the next takes place gradually instead of suddenly. This is the second principle of herringbone gearing, and may be termed *continuity of action.* In straight gears the continuity of action is a function of the number of teeth in the pinion. Straight pinions with less than twelve teeth are seldom made, and more than that number must be used if the drive is to be even moderately satisfactory.

23 In herringbone gears continuity depends on the relationship between the face width and the number of teeth in the pinion. Pinions with as few as five teeth have been used with success merely by increasing the face width to suit such extreme conditions. This feature, which is peculiar to herringbone gears, has made practical the adoption of the extremely high ratios of reduction hitherto considered impossible.

24 The third principle of herringbone gearing is that *the bending stress on the teeth does not fluctuate from maximum to minimum* as in straight gears, *but remains always near the mean value.* This feature is of special importance in rolling-mill driving and work of a similar nature.

25 To summarize the foregoing arguments: The action of herringbone gears is continuous and smooth; there is no shock of transference from tooth to tooth; the teeth do not wear out of shape; the bending action of the load on the teeth is less than with straight gearing and does not fluctuate to anything like the same extent; the gears work silently and without vibration; the phenomenon commonly termed back-lash is absent; friction and mechanical losses are reduced to a minimum; herringbone gears can be used for higher ratios and greater velocities than any other kind.

26 These advantages are limited to gears which can be produced

with a degree of accuracy which will insure the practical realization of the principles involved.

THE PRODUCTION OF HERRINGBONE GEARS

27 Herringbone gears may be produced in a variety of ways which differ from each other as widely as the character of the product. Until a few years ago all gears of this type were molded. The limitations of molded gearing are analogous to those which would be experienced if a journal were set to run in a molded bearing. Just as the bearing would touch the shaft only in spots, so molded gears utterly fail to give the intimate contact all along the teeth which is necessary to secure the realization of true helical gear action. It is obvious that if the teeth touch only in a few high places, they will be subjected to all the evils of shock, stress, and inequality of motion which it is desired to avoid. If the gears are particularly well molded, some mitigation may be expected when they become well worn, but initial wear is accompanied by a departure from correct tooth shape.

28 For slow speeds a well-molded helical gear is no better than a straight gear with cut teeth, and for high speeds it is not as good. The natural smoothness of helical action does no more than compensate for the inaccuracies of tooth form and spacing. The modern herringbone gear must have cut teeth if its advantages are to become real.

29 Cut herringbone gears may be broadly divided into two classes, two-piece and one-piece gears. The difficulty in the way of cutting double helical teeth in a single blank gave rise to the two-piece variety. The same methods of cutting may be used for both kinds.

30 The disadvantages of the two-piece type are fairly obvious. There is the expense of two complete gears, the difficulty of assembling the gears so they are in accurate register with each other, and the necessity for very complete fixing if they are to perform hard service without getting out of register.

31 High ratios, perhaps the strongest feature of the one-piece type, are not within the scope of the built-up gear, because the pinions must be assembled on a separate shaft and the pitch line must be far enough from the surface of the shaft to allow room for the necessary bolts or rivets used in fastening the two portions together.

32 The one-piece pinion, however, may be cut solid with its shaft, so that its pitch diameter need be but very little larger than the latter.

33 The known methods of cutting helical gears may be divided

into four classes: (*a*) milling by formed disc cutters; (*b*) milling by end mills; (*c*) generating by shaping or planing methods; (*d*) generating by hobs.

34 Milling by formed disc cutters is unsatisfactory because, in addition to the usual errors of step-by-step division, there is the difficulty of making the cutters to the *normal* tooth shape with sufficient accuracy to insure correct circumferential shape for the gears cut. This difficulty is increased by reason of the fact that a disc cutter cannot cut its own shape in a spiral groove. Let it be noted that the cutters must be formed empirically, that their number must be very large to meet the requirements of a general gear business, and that the accuracy of each gear turned out depends on the combined efforts of the toolmaker and draftsman who produced the cutter. Worst of all, two different cutters must be used for a gear and pinion. This method will produce indifferent herringbone gears which may be built up with teeth in register or made in one piece with staggered teeth.

35 The use of end-mills is open to all the objections given with regard to disc cutters, with the single exception that the cutter does leave a fair approximation to its own shape in the groove which it cuts. The end-mill has a host of disadvantages peculiar to itself which render it even less efficient than the disc cutter for general work. In the first place, it is a small tool with very little wearing surface and no capacity for dissipating the heat generated at its cutting edges. The great variation in diameter between point and base renders it difficult to arrive at a cutting speed which will satisfy the conditions at both ends of the cut. When used for any but the largest pitches, outside the range of general gear practice, the mills quickly become clogged with cuttings, overheat and burn. A burnt end-mill, clogged with cuttings, produces a wider groove than a good mill. This is a fruitful source of spoiled work. To complete one fair size gear by the end-mill process requires quite a number of cutters. This not only makes the expense heavy, but must necessarily result in an inaccurate gear.

36 Every cutter used must be formed to gage and hardened. After being hardened, it will run out of true a little in most cases, thereby cutting a shape different from that for which it was designed. In end-mill gears it is not merely a case of getting accurate conjugate tooth shapes in gear and pinion made with different cutters, but the teeth in a single gear may have a dozen different shapes. The process is so slow that it cannot compete with other methods, quite apart from the doubtful quality of the gears produced.

37 End-milled herringbone gears are usually made in one piece with the teeth joined at the center. Since the cutter is shaped to the *normal* pitch, it follows that, in changing over from right to left-hand helix, it leaves a thick wedge in the center of the face that must be removed by a subsequent operation. The teeth of end-milled herringbone gears do not bear over the center portion.

38 Generating processes by the shaping and planing types, while successful for straight-cut gears of relatively small size, are not used to any extent for large diameters or heavy pitches. The reason for this may be found in the nature of such processes. The gear blank is required to make a quick angular movement after each stroke of the cutting tool and to come to rest again before the next stroke. Such methods are difficult to apply to large gears on account of the inertia of the gear blank and its support and the consequent difficulties of controlling the short intermittent movements. These difficulties are much increased when such methods are applied to cutting helical teeth because the blank must make definite and rapid angular movements during each stroke in addition to the motion between strokes. No machine has been devised which will satisfactorily deal with the problem on these lines.

39 The hobbing process as applied to straight-cut gears has proved so successful as to arouse a storm of adverse criticism from those who are interested in other methods of gear production. It is not difficult to understand why this process has sprung into prominence in a comparatively short time. It is essentially a rational process. The shape of the teeth is generated from spiral hobs, the threads of which are cut to a plain rack section. There is nothing empirical about a hob; it is a straightforward thread cutting, gashing and relieving proposition. One hob will cut any gear or pinion of one pitch. This feature alone eliminates a host of errors which are characteristic of gears produced by milling methods. The hob revolves continuously while cutting, as does the gear blank. The feed is also continuous. There are no cutting and return strokes, no intermittent starting and stopping of gear blanks, as in other generating processes. These features do not necessarily insure the production of accurate gears, but they offer greater facilities to the designer for the achievement of the desired result. The hob is a substantial tool with plenty of wearing and cooling surface, it can be made to stand up to very fast production and to last for a long time. The continuous nature of all motions used in hobbing a gear blank enables this process to be used for the production of the heaviest

gears, there are no limitations such as are encountered in other processes. The limit to the size of a hobbing machine is set by the dimensions of the largest gears which are required in sufficient quantities to pay for the investment. There are no technical limits whatever.

40 Nevertheless, there are some slight defects in the hobbing process as applied to the production of straight-cut spur gears. A hob is a worm thread, and as such must have a spiral angle depending on the relationship between the pitch of the thread and the diameter of the hob. A straight-cut gear has no spiral angle, hence the spiral hob must be inclined, more or less, to bring the cutters in line with the tooth spaces to be cut. In order to cut correct teeth, the axis of the hob should be perpendicular to the axis of the gear blank. In such case the hob will generate involute teeth if its threads are cut to the same axial section as the straight-sided parent rack for the required pitch. Since the hob must be inclined to cut a spur gear, the teeth are not generated from the axial or rack section, but from a diagonal section. The axial pitch of a hob for cutting spur gears is not the same as the pitch of the gears, which it cuts. The normal pitch of the hob threads must be the same as the gear pitch.

41 Hobs for cutting straight spur gears are usually made of large diameter to reduce the spiral angle and consequent errors of tooth form to a negligible minimum. As a natural consequence, such hobs have only one thread, while their large diameter requires a slow speed of rotation to keep the cutting speed within proper limits. The effect of this is that the blank revolves very slowly, and a coarse feed must be used to keep up the output.

42 It is one of the peculiarities of hob action that only one tooth of the hob puts the finishing touch to the bottom of a tooth space once in each revolution of the gear blank. If the feed is coarse, there will be noticeable feed marks and roughness in the gear teeth produced. A coarse feed used with a hob of large radius throws severe stresses on the hob arbor and its supports.

43 The necessity of a swivel motion on the hob slide, to enable straight spurs to be cut with hobs of varying spiral angle, compels the use of a hob drive which passes through the pivot. It is almost impossible to design such an arrangement without undesirable restriction in the dimensions of driving gears and shafts combined with excessive overhang of the hob arbor in relation to its supporting slide.

44 The general want of rigidity about most hobbing machines used for the production of spur gears, is traceable to the above causes.

Rational critics of the hobbing process have based their objections on these features.

45 The hobbing process properly applied to the production of herringbone gears has none of the disadvantages incidental to its application to spur gear cutting, which have been shown to lie in the necessity of inclining the hob axis. Since a helical gear and a hob must both have a spiral angle, it is only necessary to make the thread angle of the hob the complement of the corresponding angle of the gear teeth to secure the advantages of perpendicular fixed axes. These are of great practical value. Since the hob axis is always perpendicular to the axis of the gear blank, it follows that the teeth are

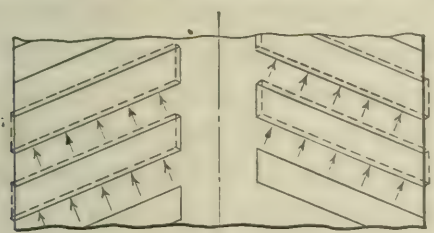


FIG. 3

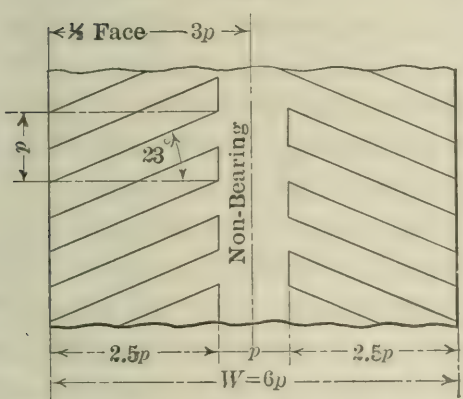


FIG. 5

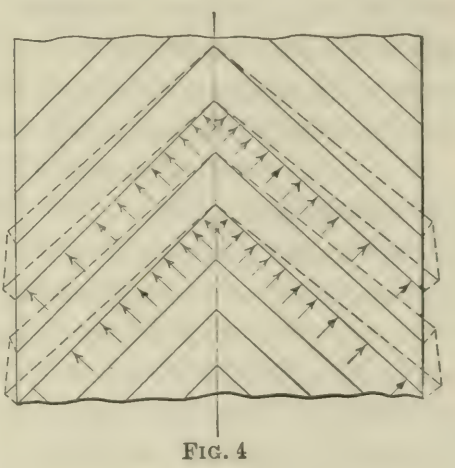


FIG. 4

FIGS. 3 TO 5 DIAGRAMS SHOWING TOOTH PRESSURES AND ANGLE NECESSARY FOR CONTINUITY OF ACTION

generated from the axial and true rack section of the hob, while the linear pitch of the hob is the same as the circular pitch of the gear which it cuts. The hob axis is fixed and the hob can be supported on a rigid slide with the minimum of overhang. There is no restriction to the size and strength of the gears and shafts used to drive the hob.

WUEST HERRINGBONE GEARS

46 It was explained in Par. 2 that the teeth of the Wuest gears are so designed that those on the right and left-hand sides of the

gear are stepped half a space apart and do not meet at a common apex at the center of the face, as in the usual type of herringbone gear. It has often been argued that the ordinary herringbone tooth is stronger than the Wuest tooth, because the latter lacks the support given by the junction of the teeth at the center. This argument would be sound if gear teeth were ever stressed to anywhere near their breaking point. But it has been found in practice that considerations of wear so far outweigh those of mere breaking strength that a gear which is designed to give reasonable service will carry anywhere from ten to twenty times the working load without fracture. A point of vastly greater importance is that the stepped form will wear more evenly under extreme loads than the ordinary type. The reason for this is shown in Figs. 3 and 4. The resultant tooth pressure is always normal to the teeth and tends to bend them apart. The stepped form offers a uniform resistance along its whole length, carrying the load from end to end (Fig. 3). The teeth of ordinary herringbone gears tend to separate more at the sides than near the supported center, causing the load to be concentrated toward the center (Fig. 4).

47 Any system of gearing, if it is to be generally applied, must be interchangeable. The variable features of involute spur gear teeth are limited to the pressure angle, addendum and dedendum. In a herringbone-gear system, we must have, besides, uniformity of spiral angle and relative position of the right and left-hand teeth.

48 The standards which have been adopted for Wuest gears are the result of experience gained in Europe during the last six years. The spiral angle of the teeth is about 23 deg. with the axis. The choice of this angle is controlled by a number of considerations, the most important from the user's standpoint being that the angle must be sufficient to allow the engagement of successive pinion teeth to overlap within a reasonable face width. Once this condition is satisfied, there is no advantage in an increase of spiral angle, while there are disadvantages in the use of steep angles. It was necessary, before choosing a definite spiral angle, to determine what constitutes a reasonable face width for this class of gearing.

49 Since the nature of the action eliminates shock, it follows that the pitch required for given conditions will be much finer than would be chosen for spur gears. On the other hand, the face width will not be less, because there is as much necessity for wearing surface with one kind of tooth as with the other. Spur gears are usually made with face width equal to three or four times the pitch. Herringbone gears may conveniently have a face width equal to six times the pitch

not because the width of this type need actually be greater, but by reason of the pitch being proportionately less.

50 Starting with a width equal to six times the pitch, and allowing once the pitch as the non-bearing portion in the center, there remains two and one-half times the pitch available for the teeth on each side. To insure continuity of engagement under all ordinary conditions, each tooth is inclined so as to cover an advance of once the pitch within its length. The angle of 23 deg. satisfies this requirement (see Fig. 5). There are a few cases where an angle less than 23 deg. would be sufficient, a steeper angle is only needed if the available face width has to be unduly restricted. Neither of these extreme conditions should influence the choice of angle for an interchangeable system best adapted for general use.

51 There are other good reasons why a moderate spiral angle is to be preferred. In all spiral gears the pressure acts in a direction normal to the teeth and is the resultant of the tangential (driving) and axial pressures. The normal pressure becomes greater in proportion to the useful driving pressure as the spiral angle is increased, while the available normal tooth section becomes less (Fig. 6). When the spiral angle is sensibly steeper than the angle of repose for the materials in contact, there is a tendency for the teeth to bind with a wedge action. Herringbone gears with abnormally steep spiral angles show loss of efficiency and increased wear from this cause.

52 The pressure angle which has been adopted for standard gears is 20 deg. The teeth are shorter than the usual standards, with addendum 0.8 and dedendum 1.0. These standards of tooth height and pressure angle have been adopted after systematic trials and experience extending over several years of regular manufacture. The high ratios used with these gears call for an average pinion diameter which is less than is used with straight spur-gears for similar duty. The teeth are generated by hobs, and the short addendum combined with wide angle gives satisfactory tooth shapes, without undercutting of teeth, for small pinions. Pinions with very few teeth are cut on the well-known system of enlarged addendum which is used for small wormwheels and bevel pinions. The teeth are cut to diametral pitch standards, measured circumferentially as with ordinary spur gearing.

53 The dimensions proposed for an interchangeable system for these gears are as follows:

Tooth shape ..... Involute
Pressure angle ..... 20 deg.
Spiral angle ..... 23 deg.

$$\text{Pitch diameter (20 teeth and over)} = \frac{\text{Number of teeth}}{\text{D.P.}}$$

$$\text{Blank diameter (20 teeth and over)} = \frac{\text{Number of teeth} + 1.6}{\text{D.P.}}$$

$$\text{Pitch diameter (under 20 teeth)} = \frac{0.95 \times \text{Number of teeth} + 1}{\text{D.P.}}$$

$$\text{Blank diameter (under 20 teeth)} = \frac{0.95 \times \text{Number of teeth} + 2.6}{\text{D.P.}}$$

Addendum ..... $\frac{0.8}{\text{D.P.}}$

Dedendum ..... $\frac{1.0}{\text{D.P.}}$

Full depth ..... $\frac{1.8}{\text{D.P.}}$

Working depth ..... $\frac{1.6}{\text{D.P.}}$

Standard face width for gears with pinions of not less than 25 teeth ..... 6 times circular pitch
Face widths for high ratio gears with small pinions ..... 6 to 12 times circular pitch

54 When a pinion of less than 20 teeth is used with a standard gear, the center distance must be slightly increased to suit the enlargement of the pinion. If it is desired to keep the center distance to the standard dimensions, the gear diameter may be reduced by the amount of the enlargement given to the pinion. For example: If a pinion of 10 teeth, 5 D.P. is to mesh with a gear of 90 teeth at 10 in. centers

$$\text{Pitch diameter of pinion} = \frac{0.95 \times 10 + 1}{5} = 2.1 \text{ in.}$$

$$\text{Enlargement over standard pinion} = 0.1 \text{ in.}$$

$$\text{Pitch diameter of standard gear} = \frac{90}{5} = 18.0 \text{ in.}$$

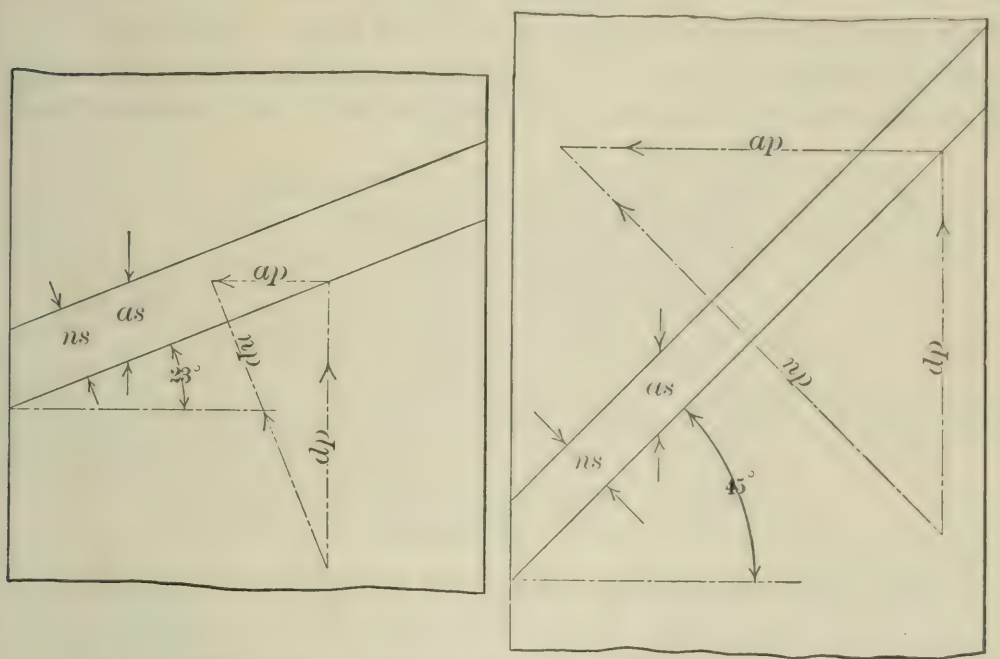
$$\text{Reduced pitch diameter of gear} = 18.0 - 0.1 = 17.9 \text{ in.}$$

$$\text{Center distance} = \frac{17.9 + 2.1}{2} = 10 \text{ in.}$$

Strictly speaking, there can be no enlargement or reduction of the pitch diameter in a pinion or gear of given pitch and number of teeth.

It is convenient to assume this enlargement and reduction, while using teeth with long and short addenda but standard depth.

55 In these gears the teeth need not have the same breaking strength as with spur gears because they have not to combat the heavy and indeterminate stresses which arise from inequalities of angular velocity. On the other hand it is necessary to provide against



		23 Deg.	45 Deg.
as	Axial Section		
ns	Normal Section	$ns = 0.9205\ as$	$0.707\ as$
dp	Driving Pressure		
ap	End of Wedge Pressure	$ap = 0.424\ dp$	$1.0\ dp$
np	Normal Pressure	$np = 1.09\ dp$	$1.414\ dp$
	Relative $\frac{np}{ns}$	1.19	2.0

FIG. 6

FIG. 6 RELATION OF DRIVING PRESSURE TO ANGLE OF TEETH

rapid wear. By using a finer pitch, each tooth has less individual wearing surface, but this is more than compensated for by the larger number of teeth in simultaneous contact than with gears of equal diameters but coarser pitch.

56 In high ratio gears, using pinions of exceptionally small diameter, the pitch is finer than for ordinary ratios, but the face width is extended to give the proper wearing surface.

57 The important factor in determining the proportions of the teeth is the relationship between pitch line velocity and the per-

missible specific tooth pressure; in other words, the total tooth pressure divided by the area of all the available simultaneous contact along the teeth. Theoretically, this contact has no area since it should consist of lines without breadth. Actually, an area exists, due to the elastic compression of the teeth in contact, in a way similar to that in which an area of contact exists between a car wheel and a rail. The area of contact is indeterminate, but the specific tooth pressure is proportional to the driving stress on the teeth.

58 In order to obtain a simple rule for finding the proper dimensions, the results of experience in the matter of safe working loads

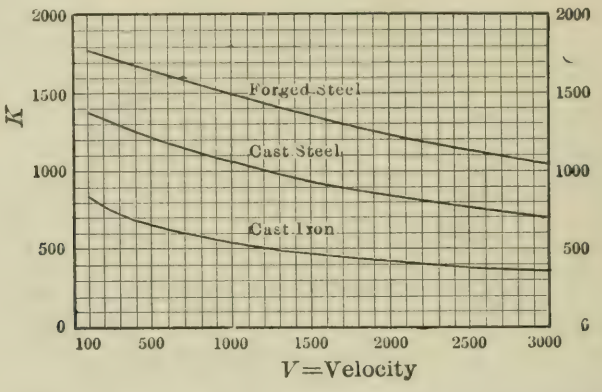


FIG. 7 SHEARING STRESS IN RELATION TO PITCH-LINE VELOCITY

under given conditions have been reduced to a relationship between pitch line velocity and the shearing stress on the pitch line thickness of an imaginary straight tooth, assuming only one tooth in engagement at a time. The shearing stress is a measure of the specific tooth pressure, and the relationship referred to affords a convenient means of arriving at reliable dimensions. The curves, Fig. 7, give values of shearing stress K in pounds per square inch on pitch line section of an imaginary single tooth for corresponding pitch line velocities V in feet per minute. The values are entirely empirical, but they are based on the results of extended experience, and lead to dimensions which are safe and reliable. Different curves are given for different materials, and it is necessary to use that curve which corresponds to the lowest grade material of the combination. The dimensions of gears can be derived from the curves in the following manner:

h.p. = brake horsepower transmitted
N = revolutions per min.
D = pitch circle diameter, in.
p = circular pitch in inches (use nearest diametral pitch).

W = total width of face, in.
V = pitch line velocity, ft. per min.
P = total tooth pressure at pitch line, lb.
K = stress factor (from curve)

$$V = \frac{\pi DN}{12} \quad \ldots\ldots [1]$$

$$P = \frac{\text{h.p.} \times 33.000}{V} \quad \ldots\ldots [2]$$

$$P = \frac{pWK}{2} \quad \ldots\ldots [3]$$

$$P = 3p^2K \left(\begin{array}{l} \text{in normal gears of moder-} \\ \text{ate rat o, and face width} \\ \text{equivalent to six times the} \\ \text{circular pitch} \ldots\ldots \end{array} \right) \quad [4]$$

$$p = \sqrt{\frac{P}{3K}} \quad \ldots\ldots [5]$$

For high ratio gears take $W = Rp$ (R = ratio to 1) up to maximum of $W = 10p$. Then

$$p = \sqrt{\frac{2.5P}{RK}} \quad \ldots\ldots [6]$$

59 Usually the values of h.p. and N are known. In many cases the diameters or center distances are fixed, and there is no choice of dimensions. When the diameters are not fixed there are many solutions to the same problem and it becomes largely a matter of experience which to select in order to obtain the most economical and satisfactory gears.

60 In normal gears it is safe to aim at pitch line velocities between 1000 and 2000 ft. per min. with 1500 ft. as a fair average. If the pinion is to be fixed to a motor shaft without external support, the diameter must be greater than when it can be supported on both sides. Cast-iron is preferable to cast-steel for gears of large diameters and moderate power, but the latter will be found more economical for high tooth pressures. Pinions are usually made from steel forgings of 0.40 to 0.50 per cent carbon. Soft pinions should never be used for herringbone gears. Besides being bad engineering practice they are unnecessary, because steel pinions run without noise and last much longer.

61 The following is a typical instance of the range of choice in dimensions: A pump which requires 150 h.p. at 50 r.p.m. is to be driven from a motor at 500 r.p.m. with shaft end $4\frac{1}{2}$ in. in diameter.

If the shaft is unsupported, it is not desirable to use a pinion of less than 10 in. If the shaft is extended to a third bearing a $7\frac{1}{2}$-in. pinion can be used. If the pinion is cut solid on its shaft and coupled to the motor, its diameter can be reduced to 5 in. The three arrangements work out as follows:

		Material for Gear	V	P	K	Diametral Pitch	Face Width
A	10 in. and 100 in...	...Cast-iron	1300	3800	500	2	$9\frac{1}{2}$
B	$7\frac{1}{2}$in. and 75 in.	.....Cast-iron	975	5100	530	2	12
C	$7\frac{1}{2}$in. and 75 in..	....Cast-steel	975	5100	1060	$2\frac{1}{2}$	$7\frac{1}{2}$
D	5 in. and 50 in.	.. .Cast-steel	650	7600	1150	$2\frac{1}{2}$	$12\frac{1}{2}$

Any of the above gears will do the work satisfactorily. C is the most economical, but B or D would make the least noise. If a gear case is to be provided then D will give the most economical combination.

62 The foregoing data can be used for finding the required dimensions of herringbone gears for all general applications. In most cases it is sufficient to calculate the tooth pressure from the average working load. When the maximum load is very far in excess of the average, it is usual to take a mean value between the two. Gears for electric mine hoists and single throw pumps fall within this category. Machine tools, when driven from variable-speed motors, are required to perform maximum duty at minimum speed only for short periods at long intervals. It is sufficient when getting out gears for a drive of this kind to reckon with the rated output of the motor at the mean between its maximum and minimum speed.

63 There are two special cases where the ordinary methods of calculation should not be used. Rolling-mill gears are subjected to stresses which are so far in excess of the average working load that it is necessary to consider carefully the strength of the teeth in regard to possible overloads. Extra high velocity gears, as such are used for steam turbines, require additional wearing surface and are characterized by extreme width of face combined with abnormally fine pitch. These are two extremes in gearing and their design is best left to those who have made a special study of them.

64 Before describing some special applications of these gears to the needs of various industries and machines, it may be of service to summarize the salient features of the gears and the changes of viewpoint which these features have engendered.

65 The smooth and continuous action is virtually independent of the diameter or number of teeth in the pinion. Extremely high ratios

of reduction can be used without fear of uneven driving or undue wear and without need for unwieldy gear diameters which would be disproportionate to the general design. High ratio gears of this type transmit power with practically no more loss than low ratio gears. They are far more efficient than belts, ropes, worm-gears or compound trains of spur gearing, while their adoption results in a wholesale reduction of countershafts and bearings which reduces the power consumption and running costs to a remarkable degree.

66 There are many instances where spur gears cannot be used because the vibration which they set up has a detrimental effect on the driven machine or its product. The inconvenience of a cumbersome system of belts or ropes has usually to be borne in such cases, but it is not too much to say that the requirements of almost all of them are fully satisfied by this type of herringbone gears.

67 The application of spur gears has been much restricted by the noise which they make when run at high velocities. The use of rawhide or other soft materials has proved only moderately successful for comparatively light work and is quite unsound for heavy gear practice. Herringbone gears in combination with durable steel pinions make less noise than soft pinions and spur gears when new, and while the former become quieter with use, the latter soon begin to rattle as they become worn. It should be noted that the use of soft pinions, while mitigating the nuisance of excessive noise, does not reduce vibration or unevenness of motion.

68 There is a limit to the pitch-line velocities at which spur gears can be operated beyond which it is unsafe to use them. This limit is far below the minimum velocities which can be used in connection with steam turbines of economical design and high power.

69 Accurate herringbone gears operate quite smoothly at velocities which are impossible for other types. This feature would appear to reserve for them a field of application which has great possibilities and is likely to cause some great changes in the standard practice of today.

APPLICATION TO STEAM TURBINES

70 There are many instances when the power of the turbine can be more conveniently applied in mechanical form than through electrical transmission. The advantages of high-power turbines have been discounted in many instances by the necessity for expensive electrical outfits with their attendant losses of generation, distribution and conversion.

71 Direct-connected steam turbines for marine propulsion have been only partially successful in a very limited field. The screw propeller, working in a dense medium, has an economic speed of rotation which is far below the best speed for a steam turbine of corresponding power. It is only when the power required is very great and the speed of the vessel unusually high that the direct-connected turbine

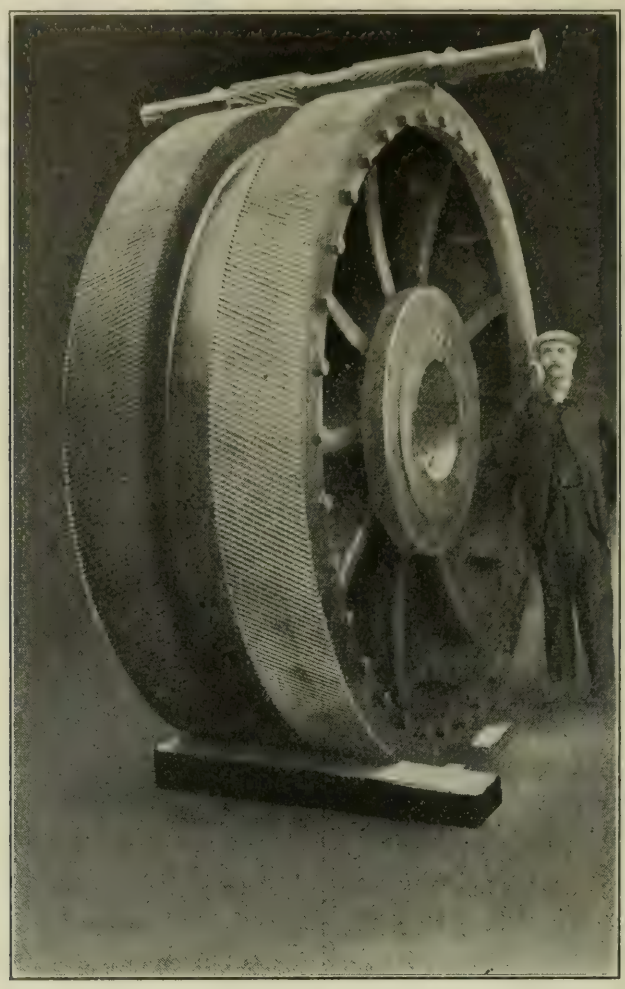


FIG. 8 GEAR AND PINION FOR 500-H.P. MARINE TURBINE

can be applied, and even then the application does not do full justice to either turbine or propeller, while the first cost is much higher than it need be. The use of direct-coupled turbines is confined to ocean fliers and ships of war. Ordinary vessels of commerce, which are built in vastly greater numbers, cannot be adapted to turbine power in this form. Mr. Parsons attacked the problem of applying the turbine to an ordinary freight steamer of moderate power. To

this end he purchased the S. S. Vespasian, a modern tramp with triple-expansion engines of about 1000 h.p. and a speed of 11 knots with propeller running at 75 r.p.m. As a preliminary to the installation of geared turbines on this vessel, the original engines were overhauled and tuned up and a series of coal consumption trials made under regular sea-going conditions.

72 The engines were then removed and for them were substituted a pair of steam turbines connected to the propeller by the herringbone gears illustrated in Fig. 8. Each turbine develops about 500 h.p. at 1500 r.p.m. The propeller runs at the original speed of 75 r.p.m. Each turbine is coupled to a herringbone pinion with teeth cut solid on a shaft of soft-grade chrome nickel steel. The two pinions mesh with rolled-steel gear rings mounted on a cast-iron spider which is keyed to the propeller shaft. The whole gear system is enclosed in a case, and the teeth are kept lubricated by oil jets. The great width of the pinions in proportion to their diameter made it necessary to provide room for bearings between the right and left-hand teeth. The proportions of this remarkable gear unit are as follows: Pinions, 20 teeth; gear, 398 teeth, 4 diametral pitch; teeth of involute form, 20 deg. pressure angle, 23 deg. spiral angle; over-all face width, 34 in. including 10 in. space for bearing; actual face width, 24 in.; ratio of reduction, 19.9 to 1.

73 This gear has now been running regular voyages for more than a year and has covered over 20,000 miles. The results have been interesting and satisfactory. The efficiency of the gear is fully 98 per cent, including the losses in the bearings on the gear case. The geared turbine shows a sustained all-round saving in fuel consumption of more than 25 per cent over the original engines. The gear runs with remarkable smoothness and without noise or vibration. The wear on the teeth is negligible after 20,000 miles, being only 0.002 in. at the pitch lines of the pinions. Even this small wear is nearly all traceable to inadequate arrangements for freeing the oil from grit during the first runs.

74 Not by any means the least important gain is in the behavior of the vessel in rough water. There has been no racing of the propeller under circumstances where this disagreeable feature was painfully evident with the original installation. This is due to the high rate at which the turbine motors store up energy with change of speed, which makes it impossible for a large change to occur during the time when the propeller is partially uncovered. Such a record as this is so conclusive that the use of geared turbines for marine pro-

pulsion must shortly become general. Herringbone-geared turbines aggregating 3600 h.p. have recently been fitted to the U.S.S.S.Neptune by the Westinghouse Machine Company. The performance of this vessel will be awaited with interest.

75 Mr. Parsons has made a successful introduction of herringbone gears in combination with a steam turbine for driving a continous plate mill. This is a proposition which few engineers would have considered seriously, yet it has proved a success from every point of view.

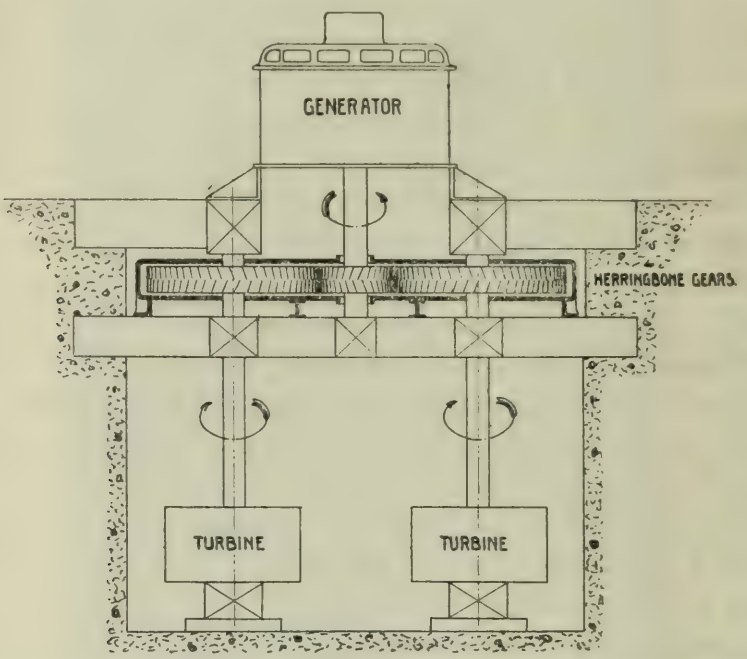


FIG. 9 DIAGRAM OF GEAR CONNECTION BETWEEN HYDRAULIC TURBINES AND AN ELECTRIC GENERATOR

76 The mill is of the three-high type with rolls 28 in. by 84 in. running at 70 r.p.m. The turbine is designed for mixed pressure, running at 2000 r.p.m. with exhaust steam at 16 lb. absolute or live steam at 60 lb. absolute and giving 750 b.h.p. The turbine is coupled to the rolls through a double train of herringbone gears. The first train consists of a chrome nickel steel pinion, cut solid on its shaft and cast-steel gear. Pinion and gear have 25 and 131 teeth, $3\frac{1}{2}$ diametral pitch by 24 in. face. This train reduces from 2000 to 375 r.p.m. and is coupled to the second train through a flexible coupling. The second train includes a high carbon steel pinion of 23 teeth and cast steel gear of 127 teeth, 2 in. circular pitch by 16 in. face. This train reduces from 375 to 70 r.p.m. The final gear is overhung on the

end of a flywheel shaft 22 in. in diameter, which carries a flywheel of about 100 tons between two bearings and is coupled to the main pinion through a pair of wobblers. Both sets of gearing are enclosed in casings and are lubricated by oil jets from a pump provided for the purpose. This mill has been running without a hitch of any kind since September 15, 1910. The gears are noiseless, the installation shows remarkable efficiency, the rolls run with extreme smoothness and the pinions do not show appreciable wear.

77 The success of the geared turbine for such an application as the one described makes it certain that similar arrangements can be used with advantage for driving textile and other mills where the conditions are less severe. The geared turbine is making rapid progress for driving direct-current generators, and several large sets are now running and in course of construction.

78 Hitherto, it has been usual to couple the turbine to a high-speed alternator and to convert to continuous current through a motor generator or rotary converter. The geared turbine unit costs less money, takes up less space and has an overall efficiency at least 6 per cent better than the A.C.-D.C. combination.

79 Geared turbines have another field of application for driving centrifugal pumps. Direct driven units of this kind have poor efficiency because a compromise has to be made between turbine and pump speed which is detrimental to both. The interposition of a set of herringbone gears allows both ends to be constructed for the highest economy, and since the loss in the gears does not exceed 2 or 3 per cent there is a large all-round gain in efficiency. This applies also to turbine-driven blowers and fans.

GEARED HYDRAULIC TURBINES

80 The speed of hydraulic turbines is controlled by the available head of water supplied to them. The greater number of turbines are required to operate under low heads and must run at slow speed. Hydroelectric power has usually to be transmitted to a considerable distance and is produced in the form of alternating current of definite periodicity. The speed of the turbines may be as low as 50 r.p.m. or even less. A large direct-coupled alternator for this speed is an expensive proposition.

81 Herringbone gears can be used to speed up from the slow-running turbines to generators of normal design, speed and efficiency. The smooth action of these gears is unimpaired when the wheel drives the pinion, and high ratios of speed increase can be obtained from them

without noise and with less loss than direct coupled units will give. A typical installation of this kind is outlined in Fig. 9. This arrangement shows two slow-speed vertical turbines driving one generator.

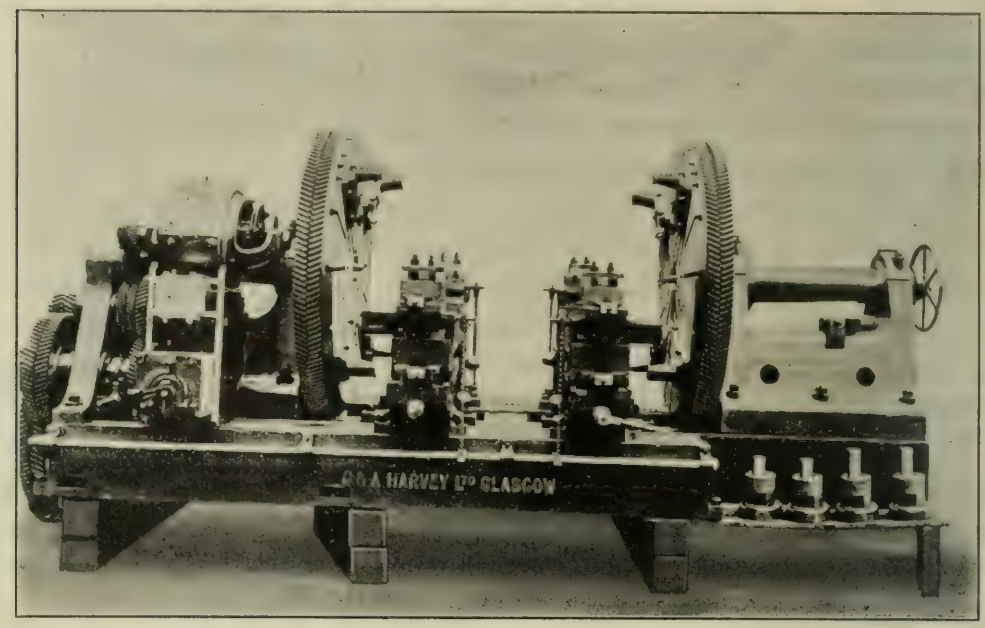


FIG. 10 TIRE-TURNING LATHE WITH HERRINGBONE GEARS

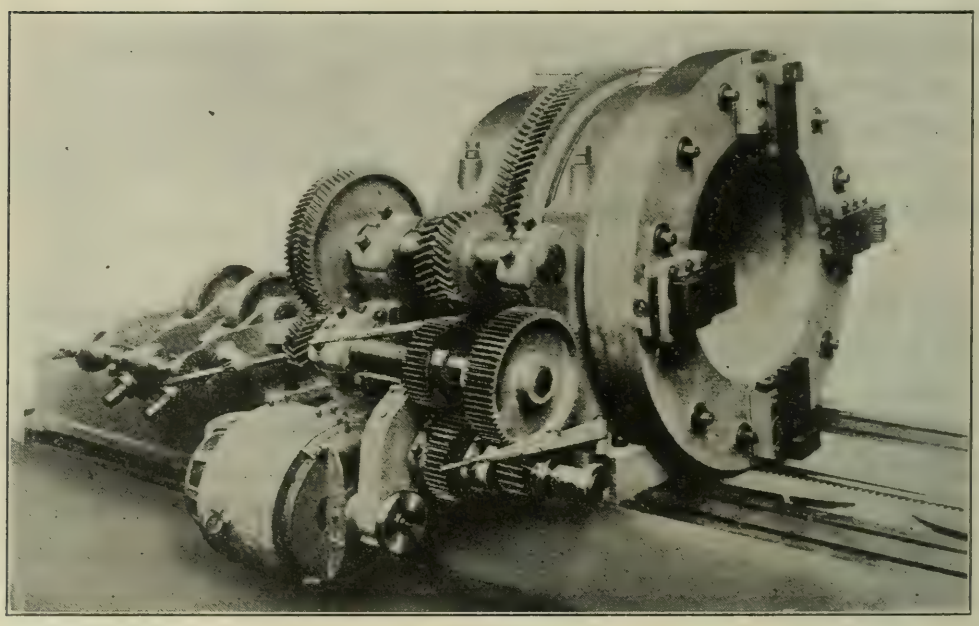


FIG. 11 GUN-BORING LATHE WITH HERRINGBONE GEARS

A herringbone gear is mounted on each turbine shaft and both gears mesh with a pinion which is coupled to the generator.

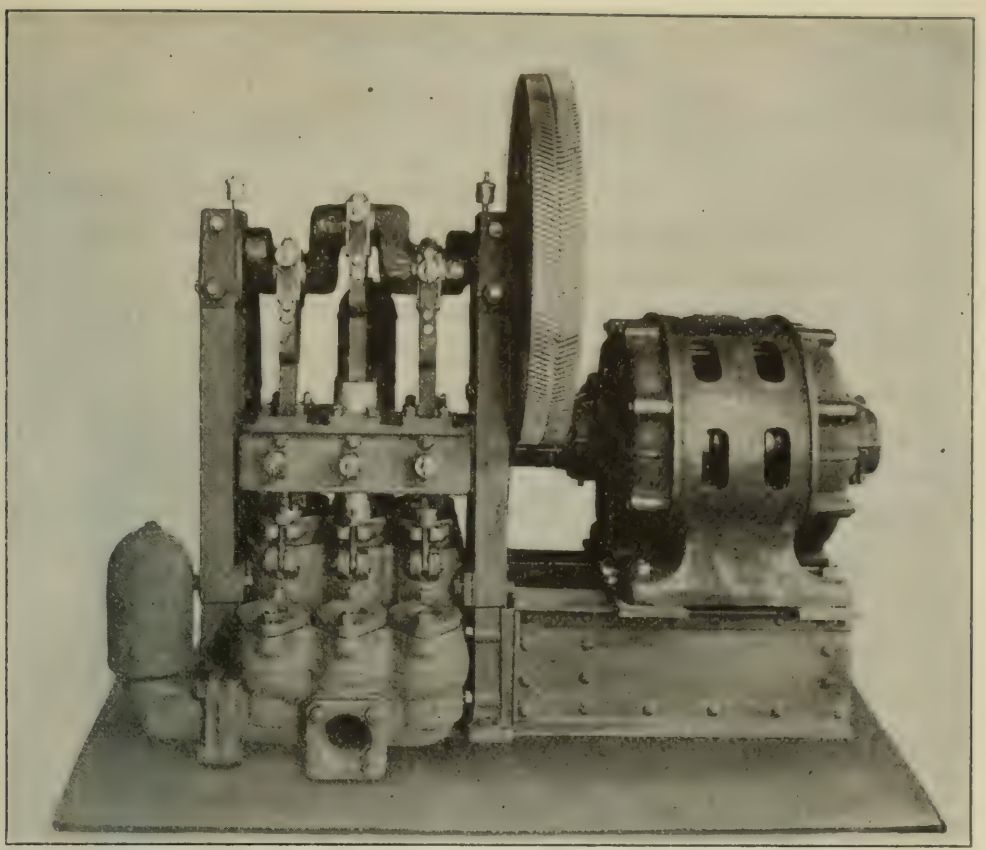


FIG. 12 HERRINGBONE GEARS APPLIED TO ELECTRICALLY-DRIVEN TRIPLEX PUMP

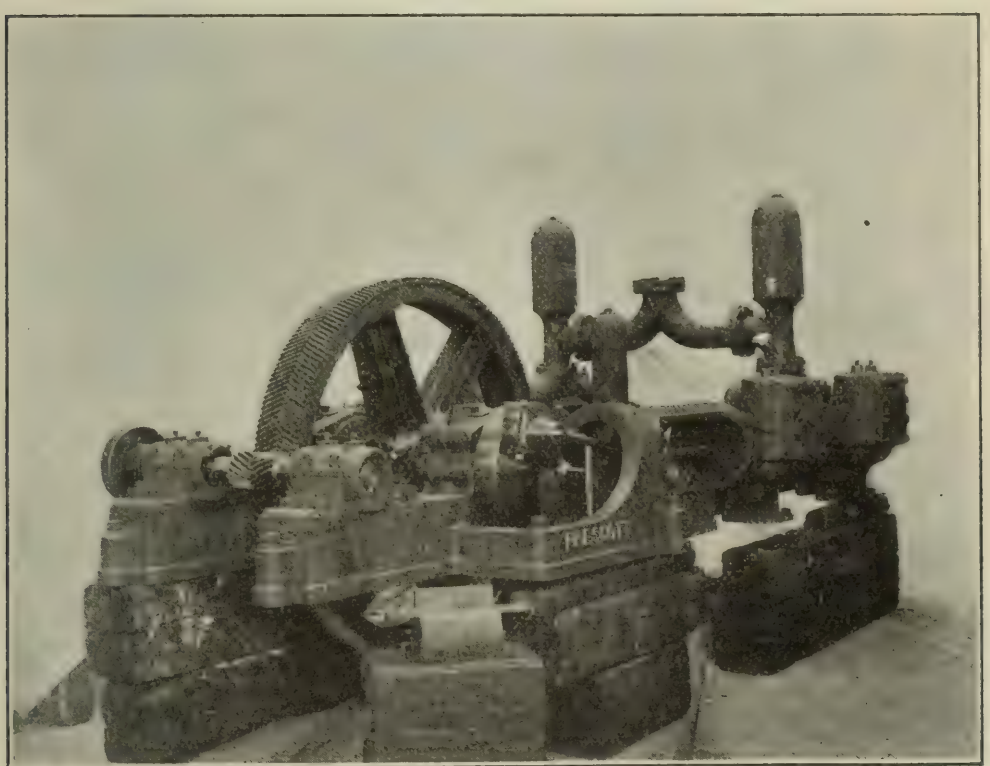


FIG. 13 GEAR-DRIVEN HORIZONTAL PUMP, GEAR RATIO 12¼ TO 1

ROLLING MILLS AND ROD MILLS

82 There are two advantages in the use of accurate herringbone gears for this class of work. The absence of shock in transmission renders breakages much less frequent than with cut spur or molded helical gears. The even transmission and entire elimination of vibration allows the finishing rolls to be gear driven for the finest work without showing gear marks on the finished product. Herringbone-geared mills run with very little noise. This may be of less consequence in rolling mills than in most other applications, but it is an improvement. Rod mills, with their quantities of high-speed

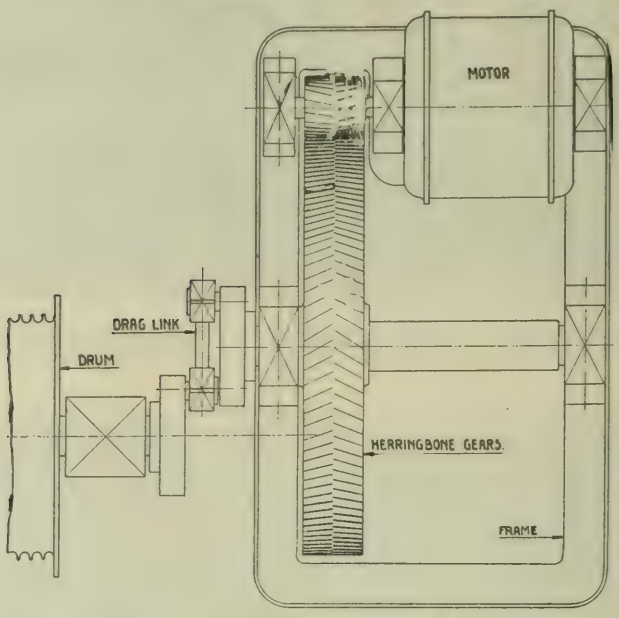


FIG. 14 ELECTRIFICATION OF MINE HOIST FORMERLY STEAM DRIVEN

gearing, can be completely transformed by using herringbone gears and mill pinions.

MACHINE TOOLS

83 The field for accurate herringbone gears in connection with machine tool driving is too extended to be considered in detail For individual motor drives this gear gives a positive transmission which is free from vibration and less noisy than so-called silent chains or rawhide pinions, while there is no trouble from slipping belts or slack chains. But the real advantage of these gears lies in the better finish that can be obtained when they are used for the entire main transmission, and in the higher output combined with reduced maintenance which they give to heavy machine tools. Chatter is eliminated. Even the speeding up to the wheels of grinding machines has

been successfully accomplished. Reversing gears for heavy planers are a revelation to those familiar only with the ordinary spur drive. Illustrations of machine tool drives are shown in Figs. 10 and 11.

PUMP DRIVING

84 Electrically-driven plunger pumps have not enjoyed the popularity that might be expected, due to the noise and vibration caused

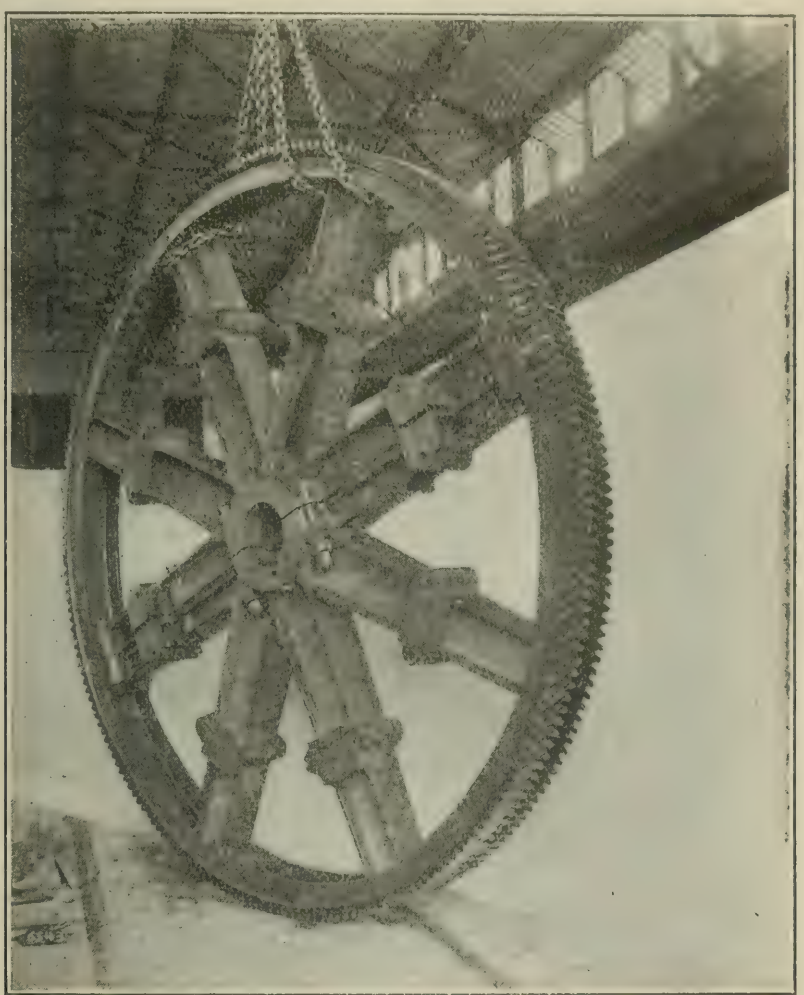


FIG. 15 SEMI-STEEL GEAR, 12-IN. FACE, 12 FT. IN DIAMETER, FOR MINE HOIST

by the gearing between motor and pump shaft. These objections are obviated by accurately cut herringbone gears, which not only give silent and vibrationless transmission but admit the use of high ratio single reductions with compact dimensions. The single reduction drive has a much higher efficiency than the ordinary double train, quite apart from the lower gear losses, because at least one countershaft can be dispensed with. Examples of such drives are

shown in Figs. 12 and 13. Similar drives are successful for air compressors and vacuum pumps which present similar difficulties to those met with in plunger pumps.

APPLICATION TO MINING

85 One of the features in the recent electrifications of the Eckstein group of mines on the South African Rand is a train of herring-

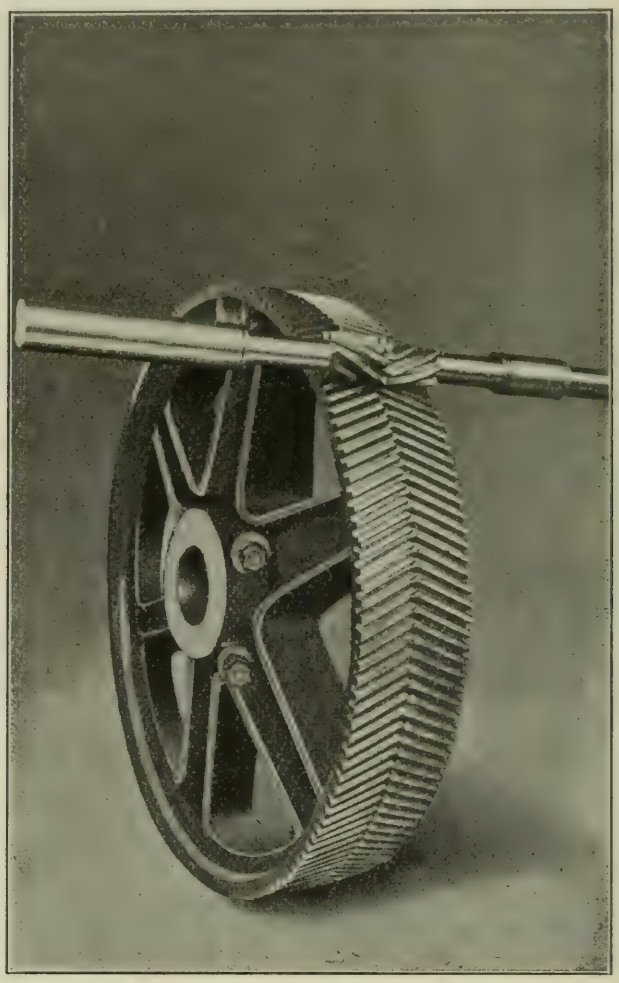


FIG. 16 TYPICAL HIGH RATIO GEAR AND PINION, THE LATTER WITH 8 TEETH

bone gears between motor and drum in the main hoists. These hoists are driven by reversing asynchronous motors; the geared countershafts being connected to the drums by drag links from the original crank pins that were used when the hoists were run by steam power. The arrangement is shown in diagram in Fig. 14. Fig. 15 shows a large gear made for mining work.

86 In installations of this type there are no slipping clutches, and

the strains on the gears are very severe, some of them having to transmit 3000 to 4000 h.p. at pitch line velocities ranging from 2000 to 3500 ft. per min. The large coal mines in Northumberland, Yorkshire and South Wales are rapidly adopting high-tension three-phase current for the distribution of power below ground. Some very large main and tail, and endless, haulages are used in these and herringbone

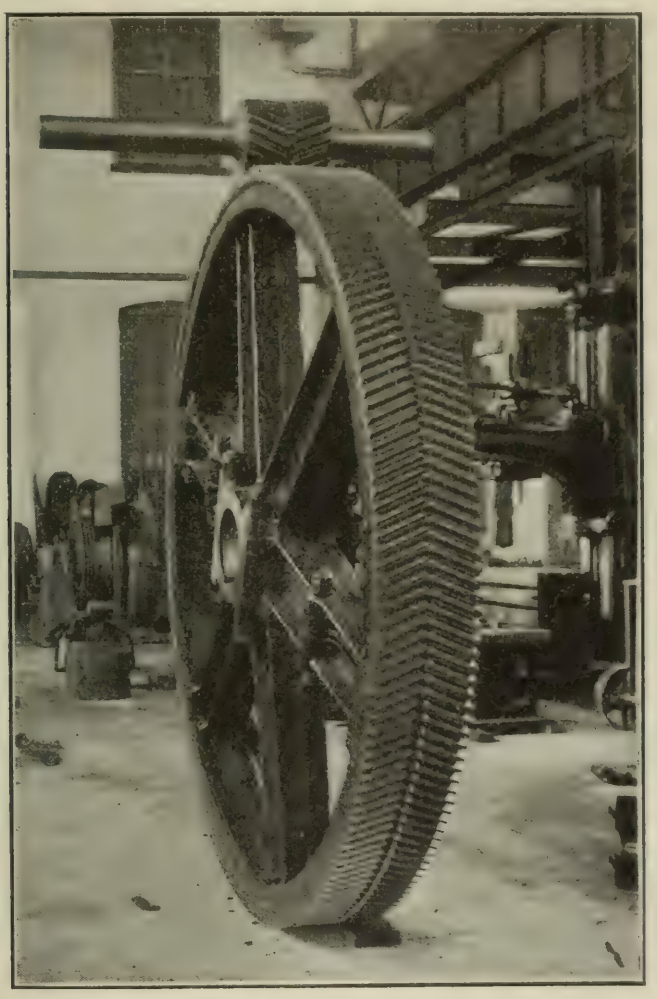


FIG. 17 HIGH RATIO PUMP GEARS, GEAR 9 FT. IN DIAMETER

gears have become a standard for this class of work. The hoists range from 30 h.p. up to 1000 h.p. and invariably use a high reduction with ratios which are sometimes as high as 15 to 1. Simplification of design and saving of space is obtained in this way, since ordinary spurs require a double train. The gain in efficiency and the absence of noise are remarkable, while the first cost of the whole outfit is often less than when cheaper gears are used. The elimination of all vibration prevents crystallization of the shafts and disintegration of the insulat-

ing material in the motors. These gears offer the same advantages for endless haulages. A higher speed motor can be used when they are adopted, while no more than two trains of gears are required.

87 An application of especial interest is for driving tipplers. These gears for this purpose have replaced worm gears because they stand up to the heavy strains without excessive wear. The available space for the gears is always limited in such cases. Some typical high ratio gears are shown in Figs. 16 and 17.

ELEVATORS

88 There are very few high buildings in Europe and the elevators there run at comparatively low speeds. As a consequence, worm gears predominate there for this class of work. The Wuest type of gear was brought out with a view to overcoming the losses inseparable from worm gears with high ratios. The needs of American sky-scrapers have caused the development of elevators in this country to run on different lines, so that the popular type of today has the rope sheave direct connected to the motor. Needless to say, the motor runs at exceedingly slow speed, usually not more than 50 or 60 r.p.m. as a maximum. Such a motor is expensive and inefficient. The system of control is wasteful to a degree. Field regulation is out of the question, and the speed control is obtained by shunting the main current through a resistance so as to reduce the volts across the armature of the motor. The survival of a type of machine so uneconomical is due to there having been no satisfactory gear system which would fit the peculiar conditions. A number of elevators are now being equipped with herringbone gears. The motor is geared to the sheave through a single train with a ratio of about 10 to 1. The maximum motor speed is about 500 r.p.m. and the speed control is nearly all obtained by simple field regulation. The following advantages are claimed: The power consumption is not more than 60 per cent of what is required for direct connected motors, the electrical switch-gear is far more simple, has less to do and is not so liable to get out of order; the motor can be repaired without interfering with the car or its suspension.

The Falk Company, Milwaukee, Wis., have acquired the sole rights for the United States under the Wuest patents and during the past year have been constructing some of the most powerful gear-cutting machinery in existence with a view to securing the required degree of accuracy in the production of gears of this type.

DISCUSSION

F. E. Rogers. The feature of novelty of the Wuest herringbone gears to gear makers and users in this country is the comparatively low cost of their manufacture. By offsetting the right-and left-hand spirals one-half the circular pitch, it is possible to cut both spirals simultaneously by the hobbing process on a specially constructed gear cutter without interference of the cutters with the teeth at the center. This obviously has an important effect on the cost of manufacture. The principal reason that the herringbone gear has not been more used in machine design, I believe, is the comparatively high cost of production when made in the common manner, that is, in two parts bolted or riveted together. It is obvious that a gear of a given diameter, face and pitch made in two parts must cost more than an integral gear of the same dimensions. Moreover, there is the well-known difficulty of cutting a right-hand and a left-hand spiral gear so accurately that when joined side by side the teeth will register perfectly. Makers of gears will bear me out in the statement that while theoretically it is easy to make a right-hand spiral of the same degree of accuracy as a left-hand spiral, or vice versa, it is not easy in practice to produce right- and left-hand mates that can be so joined as to run in perfect harmony with another pair similarly produced. Hence, there is a very great advantage commercially in being able by the Wuest principle to produce integral gears having all the valuable characteristics of common herringbone gears with none of the disabilities due to differences in the halves of each gear.

During a recent visit to the Falk plant in Milwaukee, I found that the time required for hobbing a Wuest herringbone steel gear of 9 ft. diameter and 16-in. face was about 30 hours. This is an example of the remarkable speed and adaptability of the method. It is the general opinion among gear makers that the hobbing process is adapted only to the smaller gears and pitches, but in this plant the process is applied to very large steel gears and pitches with marked economy in time of cutting as compared to other methods.

The action of a pair of Wuest gears differs in no important respect from that of other herringbone gears of the same excellence of manufacture. Hence, the principal fact brought out in this paper is the commercial advantage of the system which has made the manufacture of integral herringbone gears rapid and of low cost.

L. D. Burlingame. It is a question of manufacturing method as to how far a process of this character can be used to advantage.

Mr. Day states, in Par. 88, that high-grade and expensive machinery is required to produce gears by this process, in order to secure the required degree of accuracy. That is partly due perhaps to the fact mentioned in Par. 47, which touched upon the additional requirements of accuracy as follows: "In a herringbone gear system, we must have, besides, uniformity of spiral angle and relative position of the right- and left-hand teeth."

All of these things become matters of manufacture, and we must realize that while the herringbone-gear system has its advantages, it has also its limitations in these added requirements of accuracy. No system will be helped by trying to make it a panacea for all the ills of the appliances to which it relates or by trying to make it cover ground that is perhaps better covered by other systems. This system of herringbone gears has its own field, however, and it will be useful there in the machinery to which it is applied; but special limitations will develop if any attempt is made to carry it beyond.

The absence of back-lash is spoken of in Par. 25. I would ask what the reasoning is by which this system is free from back-lash while other systems have it. The teeth being at an angle a given amount of wear would apparently produce more back-lash than with ordinary spur gears. A satisfactory answer to this question might help clear up one of the doubts that some of us have regarding all that is claimed for this system.

F. DeR. Furman. The herringbone gear has perhaps many possibilities, but I think this paper gives more credit to its theoretical action than it is really entitled to.

Par. 19 speaks of the tendency to concentrate the load near the pitch lines when the teeth are worn away at the tips. This is, of course, true, but under these conditions there will be concentrated an excessive pressure at the pitch line. When wear occurs at the pitch line, the contact line of pressure will go back again to the tips of the teeth and will remain there until they wear down, when the pressure will again be localized at the pitch line. The advantage here, as explained, occurs only when the ends of the teeth are worn off.

Par. 22 states, "Since all phases of engagement occur simultaneously, the transference of the load from one pinion tooth to the next, takes place gradually instead of suddenly." That is true but it depends upon the teeth keeping their true form.

To obtain the advantage mentioned in Par. 19, it is necessary that the teeth be worn from their true form at the tips; to obtain the

advantage mentioned in Par. 22, it is necessary that no wear at all take place at the tips. It will be obvious that these two advantages cannot be obtained at the same time.

In Par. 24 the writer says: "The third principle of herringbone gears is that the bending stress on teeth does not fluctuate from maximum to minimum as in straight gears, but remains always near the mean value." That is going back again to the pressure contact at the centers of the teeth. When the centers of the teeth wear away the pressure causing bending stress falls again on the tips of the teeth so that the advantage here mentioned is a fluctuating one and will be more decided as the tips of the teeth wear away.

It seems to me that theoretically there is always one advantage present, whether the tooth is a perfect form for its entire length, or whether it is worn off at the tip; but all the advantages claimed are never present at any one time. There is only one theoretically correct form of gear wheels that will transmit rotation with pure rolling contact without any sliding, and that can be obtained with the herringbone gear if cut, as was once said, absolutely true in the first place and the tips of the teeth on each wheel filed off; then there will always be pure rolling and the point of contact will always be on the line of centers at the pitch point. That is the ideal, but the great objection to it is that it causes concentrated pressure and wear on the tooth surface at the pitch line.

W. C. Brown. I think I was the first to use these gears in America and, so far as I know, the only one to use the arrangement similar to Fig. 9, the difference being that I used only one wheel and drove a turbine pump instead of a generator. The ratio is eight to one, the speed of the turbine wheel being 80 r.p.m., and that of the pump 640 r.p.m. The horsepower transmitted is 350. The wheels were cut in England on the first Wuest machine before Mr. Day had constructed the improved machine, but they were not very well cut. We put in micrometer adjustments for the purpose of keeping them in a perfectly horizontal plane, but even then they were quite noisy. Mr. Day very kindly came to our assistance and explained that it would be necessary for us to float the pinion. We did this, so that it could find its own plane, using a spring for end counterbalance, and since that time the gears have been running very well.

These gears are not quite silent, but it is possible to hold ordinary conversation in the room while the machine is in operation, and I

think, if the gears were as accurately cut as they could be on the new machine there would be practically no noise. They have been in operation now about a year and a half and we have had no trouble.

THE AUTHOR. In order to answer Mr. Burlingame's inquiry as to why the herringbone gear system is free from back-lash, it is necessary to arrive at some agreement regarding the use of this term. Some confusion appears to exist between running clearance and back-lash.

A certain amount of freedom is always desirable between the teeth of a driving pinion and driven gear no matter whether the teeth are helical or straight. What constitutes a reasonable minimum of clearance in a new set of gears depends, to a great extent, on the accuracy with which the teeth are cut; but this initial clearance is not important because it must necessarily increase as the teeth become worn by use.

Back-lash, in the writer's opinion, is not to be confused with running clearance, but is rather the effect of this clearance in combination with the small, but rapid variations in angular velocity which are well known to occur between a worn spur gear and pinion. This effect is most objectionable when spur pinions with few teeth are used; it is the root of nearly all gear troubles and the reason why spur gears are unsuitable for high velocities and high ratios of reduction. A small spur pinion drives a gear with a series of impulses or blows, and the side clearance between the teeth allows the inevitable recoil to produce a chattering effect. It is this effect which should be defined as back-lash. With a properly constructed herringbone gear and pinion the latter drives with a steady and even pressure without any chatter, hence there is nothing in the action of these gears which can be defined as back-lash, if the writer understands the term correctly.

The writer recently witnessed some trials of a Wuest gear and pinion connecting a steam turbine and generator. The load was approximately 300 h.p. at a pitch line velocity of 5500 ft. per minute. The pinion had 21 teeth and both shafts were mounted in rigid bearings. There was no vibration noticeable under these conditions, yet the amount of running clearance allowed was more than usual, in anticipation that the gears would get warm and expand.

Professor Furman has spoken of unstable conditions as regards the concentration of tooth pressure and consequent wear on the teeth. No such difficulties have come under the writer's notice during seven

years' close observation of herringbone gears working under the most varied conditions.

When herringbone gears are new the uniform distribution of the load over the whole working tooth surface causes more wear at the ends, where the sliding is greatest, than near the pitch lines, where the sliding is least.

The consequent wear that takes place does not concentrate all the load at the pitch line, but merely redistributes the load so that each portion of every tooth carries just as much in proportion to the sliding that takes place as is needed to equalize the subsequent wear over the whole tooth surface. This condition, when once established, is stable and there appears to be no reason why any further change of tooth shape should take place.

The total change from the original tooth shape is so small that the difference is practically immeasurable, assuming the teeth to be pure involutes in the first instance.

It is the writer's practice to age all new gears artificially by runing them together with a fine abrasive after they leave the cutting machines. This method could not be so successful as it has proved unless the conditions were substantially as described.

The writer's remarks on bending stress in Par. 24 refer only in part to the load concentration near the pitch line.

The maximum bending moment is there referred to as the average for one tooth as a whole, and takes into account that different portions of the tooth are in different phases of engagement and are loaded at varying distances from the base.

No. 1331

TOPICAL DISCUSSION ON CEMENT MANUFACTURE

At the Annual Meeting, December 1911, a session was devoted to a topical discussion on Cement Manufacture. Abstracts of the principal papers and discussions are given herewith.

THE DUST PROBLEM IN PORTLAND CEMENT PLANTS AND ITS SOLUTION

By Otto Schott,[1] New York

Non-Member

There are few branches of industry that can vie with that of Portland cement manufacture in the production of dust. Cement factories working with the wet or half-wet process are an exception to this, since dust from drying and grinding the raw materials and in burning the clinkers is not encountered. There are many sources of dust with the wet process in common with the dry, such as the drying and grinding of the coal, the grinding of the cylinders and the packing of the cement. In any case, the dust-eliminating question presents itself as an urgent problem. The dust collector has become a necessity for any cement plant. But little experience having been so far gathered in America in this particular respect, the explanations in this paper will be based on the experiments and experiences of German cement factories, in which such devices have been in operation for a long time.

[1] Consulting Engineer, Fifth Avenue Bldg.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers. The complete report may be consulted in the rooms of the Society.

Two points to be considered are:

a Dust-eliminating contrivances for depositing or beating down the dust originated in grinding, packing and conveying the materials when the temperature of the dust-filled air is normal.

b Removal of the dust from the waste gases of the drying drums and kilns, a more difficult problem owing to the high temperature and to the aqueous vapors contained in the gases.

DUST-ELIMINATING DEVICES

With the dust-eliminating devices first referred to, it is essential that the mills and conveying contrivances causing the dust be covered or jacketed as tightly as possible. It is furthermore of great importance to withdraw the dust immediately at the place of origin, hence to connect the mills and other contrivances directly with the exhauster destined to remove the dust-saturated air by suction, instead of allowing the dust first to penetrate to the outside and then attempting to cleanse the space saturated with the dust.

Every dust-eliminating plant consists in general of three principal parts, viz., the exhauster, the dust separator and the pipe conduits. Absolutely indispensable for every such plant is the exhauster which serves to remove by suction the dust-saturated air from the dust sources and to conduct the same to the dust separator; therefore the construction of this machine requires particular discrimination and care. Its volumetrical and mechanical efficiency and capacity must be as high as feasible, i. e., it must be so constructed as to remove by suction a quantity of air with the least possible consumption of power. The proper shape of the air wings and of the casings are in this device of the greatest consequence. The casings have to be so constructed as to permit the air to leave without any resistance, i. e., without any concussions or shocks.

The width of the suction aperture of the exhauster is dependent on the requisite working result relative to the air, multiplied by the degree of depression. Greater air actuating work and smaller depression necessitate a wide suction aperture, and vice versa. Considering the great number of revolutions in the exhausters, reversible ring lubricated bearings are the most advantageous.

For cleansing the dust-saturated air drawn through the exhauster by suction there are three technical methods in use: (*a*) diminution

of the velocity of the air; (*b*) filtration of the air through jute filters; (*c*) elimination by centrifugal force.

With the first method mentioned of removing the dust, the dust-saturated air is led into large chambers and thereby undergoes a diminution of speed proportionate to the cross-section of the chamber. Consequently a large portion of the dust is deposited on the bottom of the chamber. In order to produce satisfactory results, such chambers must have relatively considerable dimensions. It is not difficult to free small mills from dust by this contrivance, but for larger plants the requisite dimensions must increase according to the third power of the size of the plant. In Europe, where the cement industry dates rather far back and where the rotary kilns were taken from America, the spaces enclosed by the now disused ring kilns are suitably utilized as dust chambers, being sufficiently spacious for that purpose. The dust falling to the bottom of the chamber is removed automatically by a screw, spiral or chain conveyor. The separation of dust may be increased by constructing baffle works in the chamber which causes contact or friction with the air. Sometimes we find objects of resistance in the shape of wire-netting stands or hemp skeins suspended in the chambers, such devices causing the dust first to adhere to them and then drop to the bottom of the chamber. The greater portion of the dust may be deposited by the aid of such dust chambers, but a certain amount of dust will always be taken along with the air escaping from the chambers.

In a consideration of the dust filter, the device most used for the separation or elimination of dust, two principles are involved: (*a*) the exhauster is inserted between the source of dust and the filters so as to cause the dust-saturated air to be pressed into the filter; or (*b*) the exhauster is arranged behind the filter and draws the dust-saturated air from the mill through the filter. The latter construction has found great favor in Germany as made by the firm of Beth at Luebeck.

This dust collector consists of a number of separate sections of hose through which the dust-charged air is drawn by suction. The dust adheres to the interior surface of the hose while the cleansed air is allowed to escape to the outside. In order to remove the dust from the apparatus the different hose are knocked out alternately and in shifts at regular intervals of time, by means of an ingenious mechanism. The sucking action of the exhausters is suspended temporarily for these hose or sleeves by the automatic reversion of a flap or valve provided in the suction trunk. At the same instant the sleeves

are freed from the dust by repeated raising and sudden dropping. Besides this the atmospheric air flowing in under high pressure and entering the interior from the outside through the casings of the sleeves cleanses the latter completely. While part of the sleeves are thus being cleansed the exhauster draws the dust-saturated air by suction through the rest of the sleeve sets. Thus no interruption takes place and the dropping dust is carried away by a screw conveyor. To give an idea of the extensive application of these dust-eliminating plants, it may be stated that 20,000 sleeves with 1,600,000 sq. ft. of surface are in operation.

There are other apparatus also, constructed on a different principle, whereby the air is not drawn by suction through the filters, but is pressed in.

The third method of eliminating the dust is based on the utilization of centrifugal force in the shape of so-called cyclones. The dust-charged air is blown tangentially into a cylindrical sheet-metal receptacle. The coarser parts are thereby flung upon the sides and fall out below at the conically shaped point of the cyclone. The author does not hold a high opinion of the cyclones where they are to serve for the elimination of dust from milling or packing plants. This opinion is based on observations of such a plant in an American cement factory. The finest dust is not separated, but is whirled with the air current leaving the cyclone, into the outside air.

REMOVAL OF DUST FROM WASTE GASES

The dust-eliminating apparatus so far described is applied in the removal of air by suction at a normal temperature in raw mills, cement mills and in the removal of dust from conveying and packing contrivances. The apparatus may also be employed for the removal of the dust from the drying drums for raw material and coal. In these cases, however, two things have to be taken into account:

a The workman attending to the drying drum must keep the temperature of the escaping vapors high enough to prevent the steam therein from undergoing condensation; he has at the same time to take care that the exhaust vapors always contain sufficient steam and are not so hot as to burn or scald the filters.

b The casings enclosing the filters have to be made from sheet iron so that in the case of an ignition of the filters the fire is prevented from spreading.

Two such ignitions have come under the writer's observation, and to demonstrate how carefully these things should be observed, one of the cases will be related. A "Perfection" apparatus was installed for the coal mill, which had been placed out of operation from Saturday until Monday and the dust-cleansing apparatus had been put out of commission at the same time. The coal dust adhering to the yet uncleansed filters set the whole apparatus on fire by spontaneous ignition and caused a considerable conflagration. If the cleansing apparatus had been caused to make only one more revolution, so that all the filter sleeves would have been knocked out, this fire could not have originated.

Similar things may occur when the waste gases of the drying drums are conducted at too high a temperature through the filter sleeves. This may, however, be obviated by causing cold air from the free space to be drawn by suction at the same time, prior to the entrance of the hot gases, into the filtering sleeves, thereby decreasing the temperature. It goes without saying that in that case the filtering surface must be so much larger according to the quantity of the cold air introduced.

The greatest surety in freeing the heated drying drum vapors from dust can be attained by a dust chamber, which, however, must be well insulated in order to obviate the condensation of the steam.

These dust chambers, however, have to be very large, otherwise there will always be some fine dust carried along; and on the other hand, the cyclones, which in other respects would suitably serve for heated vapors, fail to give satisfactory results. The purification of the gases, therefore, from the drying drums has been variously effected and introduced in Europe by water or steam diffusion.

Dust elimination by the wet process will be applicable wherever sufficient water is available and where the slurry can be used. Such an apparatus for the freeing of the drying drums from dust has been in operation in Germany in two of the plants of the company with which the author was connected. The construction which has proved the better and more efficient consists of sheet-metal tubes bent serpentine fashion, at one side of which the dust enters, together with the vapors, while at the other side they escape in a purified condition into the free air. In the interior of the sheet-metal tube referred to, there are water nozzles rigidly attached, which serve to spray incessantly finely diffused water into the rising vapors. The dust deposited in the shape of slurry flows into a receptacle or reservoir provided underneath the apparatus and is carried uni-

formly by a pump to the mixing worms, which serve to feed the raw meal to the kilns, being thus burned at the same time with the raw meal. The consumption of water of such a separator amounts to about 22 gal. per min. The slurry pump has a capacity of 30 gal. per min. The ventilator that draws the gases through the serpentine-shaped tube needs but 4 or 5 h.p. for its operation. This apparatus, which was invented at a German machine factory, works excellently, provided the water nozzles are cleansed once a week.

In France a spraying system is employed with success, constructed as follows: The hot waste gases enter through a narrow tube from below into a wider tube, in which ring-shaped cells are provided in order to diffuse the gases as much as possible over the cross-section of the tube, thereby preventing them from flowing merely through the tube in the centre. At various places in the interior of the tube there are water sprays which are so fitted with small apertures or perforations as to cause the water to be diffused funnel fashion in all directions. The water that has been formed can run off through a laterally disposed pipe.

ELIMINATION OF DUST FROM ROTARY KILNS

Elimination of dust from rotary kilns is particularly difficult in America, because, in contradistinction to the European fashion, individual stacks of but little height are in use. On the other side a whole row of kilns is served by a brick smoke stack of about 200 to 300 ft. The author fails to see any advantage in the use of individual stacks. On the contrary, he is of the opinion that there are a number of reasons why the high brick stacks should be given decided preference.

The most simple contrivance for the elimination of the dust from the kilns is the dust chamber with automatic emptying device. In this case, however, the dust chamber is an absolute necessity, even if other contrivances for the purification of the kiln vapors from dust are employed. In the chambers sieve walls are preferably arranged to which the dust adheres. An attempt has been made to purify the waste vapors through filters and for this purpose particularly filtering sleeves have been produced from incombustible asbestos fiber material, but this mode has not proved a success. Combinations of dust chamber and certain after-cleansing apparatus have therefore been constructed. In these combined devices the vapors proceed through a dust chamber which is fitted with rocking or shock

sides and in which a great portion of the flying dust forced along is separated. In the rear of the dust chamber two water-dust depositors are provided, into which the vapors are uniformly distributed.

In order to render the deposit of dust in the dust chambers more efficacious, special depositors for the flying ashes have recently been constructed and patented. They are placed in the cross-section of the chamber and compel the smoke vapors to undergo a continuous change of direction when passing through the chamber. These depositors consist of pocket-like pieces of sheet metal which are disposed within the dust chamber.

These devices are in most cases sufficient when applied in connection with the high European smoke stacks. The very small amount of dust that is still forced along is carried away to great distances and is not observable in the vicinity of the factory.

ECONOMY OF DUST ELIMINATION

In order to demonstrate that enormous quantities of cement may be saved by the dust elimination in cement and raw mills, the following example from practical experience is given: The plants of the works referred to are fitted with Beth's sleeve filters with which daily experiments and tests were made. This factory has a daily production of 1000 bbl. In one day, i. e., in 23 hours, 3800 kg., or 166 kg. per hr., of cement was collected. As conditions are in Germany, where 100 kg. of cement cost about 75 cents, the daily gain is approximately $28.50. In the raw mill about 2000 kg. of meal are collected, representing a value of about $6. Hence the daily saving amounts to about $34.50, or $10,350 per year, reckoned as 300 days.

The dust-eliminating plant for this factory had involved an expenditure of $4500. For power, the annual outlay amounted to $750, and about $250 was expended for repairs. The expense of the plant therefore was covered by the savings in about 6 months. The quantities of dust that are regained in a dry state by the application of the dust filters amount to from 2 to 4 per cent of the entire output.

These results were produced in a Silesian factory by the aid of Beth's filters. A second instance of the practical working of a slag cement factory is one in which dust filters from the works of Simon Buehler and Baumann are in operation most successfully. In this factory which has an output of 16,500 tons, a yearly saving of 495 tons was effected in cement meal alone, by means of the filters.

which, at the rate of $4.50, amounts to $2250. This quantity of recovered material would otherwise have been waste material.

The gain that is open to proof is thus demonstrated in black and white. But a gain of at least the same value is involved in various other circumstances, which with a rationally applied dust-eliminating plant, offer considerable economical advantages.

Above all, it must be mentioned that the working power of the mills will be perceptibly increased if the heat produced by the disintegration of the never entirely dry limestone, cement clinker and steam is carried away.

Furthermore, by dust elimination the depositing of the dust on the machinery is avoided, whereby the wear and tear of the same is considerably reduced. It is the cement meal rather than the raw meal that produces wear on the rotating machine parts, in the manner of emery, more particularly when it has access to bearings which can never be so protected as to be entirely dust tight. There is not the least doubt that the superintendence and the tending of the machine plant are also materially facilitated in avoiding such disturbances.

In conclusion, it may be mentioned that the health and life of the laborers working in cement factories speak forcibly for the introduction of dust-eliminating plants, while they offer in addition the advantage of obtaining steady and more cheerful workers.

CONFIRMATION OF THE ADVANTAGES OF ELECTRICITY TO THE CEMENT MANUFACTURERS

By J. Benton Porter,[1] Philadelphia, Pa.

Non-Member

The process of manufacturing cement has in a comparatively short time made important advances both with regard to greater freedom in the choice of location for the plant, and in the size of the plant, so that plants having a capacity of less than 2000 bbl. per day are now the exception. Electricity, although practically used only since 1900, has played an important part in this development, and there are comparatively few plants that do not use electric motors on at least part of their equipment, while of the 20 or so new cement plants built in the last three or four years and representing over 60,000 electric h.p., all but a small proportion have motor drive throughout.

[1] General Elec. Co.

It is very difficult to make comparisons in the cost of operation of different plants as affected by the method of drive, because it seems to be impossible to find two which are operating under exactly the same conditions, and because the power consumption, or cost of power per barrel, depends really more on the type of grinding mills, the fineness of the finished product, and the physical properties of the raw material, than on the method of drive.

Some engineers prefer mechanical drive for the main grinding departments, owing to their belief that the electric drive, with its generator, transmission line, and motor with their attendant losses between the prime mover and the mills, is less economical than the mechanical system with only main belt and lineshaft. This may be true with a good mechanical transmission, i. e., one with a minimum of shafting and belts, but there are a number of compensating advantages possessed by the electrical system which make it ultimately preferable to engine drive as far as steam economy is concerned. The most important of these is that no plant operates at constant load (a number of plants work on about 80 per cent of full load), and the transmission losses in engine drive, which are practically independent of the load, from a small percentage at full load become a considerably greater proportion at the actual power produced, while with the electric system the losses are reduced much more nearly in proportion to the reduction in load.

Another point in favor of the electric drive is that steam turbines may be employed as prime movers, and turbines show a higher economy at all loads than reciprocating engines, and require less attendance and supplies. One of the greatest advantages of electric over mechanical drive is, however, its flexibility both as regards the power plant and the mill itself. The generating station may contain either two or three units of a total capacity somewhat greater than the maximum demand of the plant. Some cement engineers recommend the use of three units, each of about 40 per cent of the total capacity required, an arrangement which permits the operating engineer to keep the load on each unit very near the point of maximum efficiency, even though the plant is operating at a fraction of its rated output. This also allows adjustments and repairs to be made on any machine without crippling the plant, and makes the various departments more independent of the prime movers. The flexibility in the construction of a plant may considerably affect the future cost of operation, by permitting the location of the power station to be decided in accordance with water supply for boiler feed and condensing, ease

of handling coal and disposal of ashes. One plant erected in recent years found it convenient to locate the power house nearly a mile from the mill, and the saving in the cost of operation more than compensated for the energy lost in transmission. Finally, as important arguments in favor of electric drive may be mentioned, the longer life of electric transmissions, and the ease and accuracy with which the power used can be measured and apportioned between the different departments, the latter helping the cement engineer in locating trouble and reducing costs.

The electric apparatus best suited to meet the conditions of the cement industry consists of steam-turbine-driven generators and induction motors arranged for the individual motor drive. The induction motor is better fitted to deliver constant speed to the mills than any other form of drive, since it is not affected by heating or fluctuations in voltage.

ELECTRICAL POWER IN CEMENT PLANTS

By Frederick H. Lewis, Birmingham, Ala.

Member of the Society

A greater or less amount of electric power is found in practically all Portland cement plants. For lighting purposes, it is essential. For auxiliary power it has such great advantages and such marked economies that it may also be considered essential. To be able to place power at any point, however distant from the source of power, or however awkwardly situated with reference to this source of power, and to do this in an installation which is cheap and efficient, is an advantage which can be secured in no other way.

The operating economy of auxiliary electric power is remarkable. In a plant built some time ago by the writer, and equipped with direct current for auxiliary power machinery, the total generator rating was 375 h.p., while the sum total of motors and lights taking current from these generators for use as needed was 675 h.p. There was little trouble. Circuit breakers seldom went out, except in cases of short circuits on the line from some cause or other. It is quite true that the entire 675 h.p. was not in use at one time; but motors and lights totalling from individual records as much as 500 h.p. were carried by the generators without an overload record on the switchboard. The entire 675 h.p. was ready for use whenever needed at a very small expenditure of steam. There can, of course, be no comparison in

efficiency between a performance like this, and that of any power-transmission installation.

When results obtained from plants electrically driven throughout are compared with those from plants in which the heavy mill drives are by engine through a suitable power-transmission equipment, the case is not so clear. The conclusion is sometimes doubtful, or less general in its application.

The chief advantage of individual motor drives is the flexibility secured in the operation of a plant. Machines may run or be idle without affecting in any way the operation of other machines. This advantage still favors motors on mill drives, but it is evidently less marked than in the case of auxiliary power. The heavy, steady load of grinding machinery may be engine-driven by a power-transmission equipment, which is short, direct, and efficient, especially if the plant is of moderate size, while electrical machinery so applied is comparatively more expensive and less efficient than when applied to the incidental or auxiliary uses of the plant.

The disadvantages of electric power are:

a *Initial Cost.* In engine-driven plants power units are bought twice; once in the boilers and a second time in the engines, with one power loss. In an electric plant the power units are bought four times: in boilers, engines, generators, and motors, and there are three power losses. The cost of an electric plant complete, as compared with a steam plant for the main power drives of a cement mill only a few years ago, was about as 2 to 1. Turbines are now so much cheaper than engines, and require so much less room; and electric generators and motors are so much cheaper than formerly that present comparative figures are about 1.5 to 1.0.

b *Operating Cost.* In the writer's opinion there is no doubt that a completely equipped electrical plant will use more power coal per barrel of cement than a plant in which the main drives are engine-driven. In a plant producing 2000 bbl. per day, the coal consumption may be 10 per cent greater. As the plant grows larger and the losses from power transmission by engine drive increase, this difference in fuel cost will, no doubt, disappear; may, indeed, be found on the other side of the equation. The advantages of electricity become greater as the size of plants increases. Operating costs, including the supervision and repair of motors, are also greater in electric plants, especially if they are small.

c *Limited Overload Capacity.* The regular line of alternating current motors take in starting a very heavy current, at least twice as much as they take when running under constant load. Combine this fact with another, viz., that ball and pebble mills (as well as some other machines) take 40 or 50 per cent more power to start than to run. With a squirrel cage motor starting a pebble mill we have then a demand for current in starting which is three times that required under running conditions, and possibly more. In a small plant this may drop the voltage considerably, drop out circuit breakers and jolt the whole transmission. Even in large plants it is necessary often to dispense with circuit breakers with the starting side of motor switches, or else equip the circuit breakers with an inverse time element relay, so that the breaker will act only after a lapse of a certain length of time.

This objection is one particularly applicable to drives for heavy machinery, and is of relatively less importance, as plants become larger and the power required for individual motors becomes less in proportion to the whole amount of power on the transmission lines or at the switchboard. A so-called slip-ring motor is now available which requires less starting current, and in that respect is desirable for heavy duty in small plants. The squirrel-cage motor in itself, however, is more desirable.

In a Portland cement plant in full operation cost does not vary much from month to month. With a large or small output, the difference is dollars is not great. Hence economy of manufacture hinges very largely upon the output, the figure used as a divisor for the comparatively constant cost in dollars. The larger the output from any given plant the less the cost per barrel. Hence, the mechanical equipment, which will produce full output with regularity, may be advantageous even if the operating cost in dollars is higher.

In the writer's observation, the operating cost for power is higher both in fuel used and in labor (including the labor required for supervision of motors and their accessories, and for the repairs of these things), while operating results are good. It is rather difficult, therefore, to reach a definite conclusion upon the actual advantages or disadvantages of the electric mill.

As a general proposition for a plant of 2000 bbl. or less, a power drive for the raw and finishing mills with auxiliary electric power appears to be advantageous. For larger units the electric mill is better.

This conclusion is based upon purely mechanical considerations. In addition, there is the human element to be considered; an element which may neutralize great mechanical advantages, or surmount great mechanical disadvantages.

Undoubtedly a much larger percentage of costly mistakes have been made from unfortunately chosen electrical machinery than in any other part of the mechanical equipment of cement plants. This might be considered an additional disadvantage of electric plants. It certainly is a business hazard. Nevertheless, highly efficient electrical plants are regularly on sale and available to the discriminating.

PROTECTION OF LABORERS FROM ACCIDENTS AND INJURY TO HEALTH IN CEMENT PLANTS

By Otto Schott, New York

Non-Member

In comparison with the factory help in European countries the American laborer offers the advantage of having learned, through his independent rearing and through the extensive cultivation of sports, to move with ease and self-reliance amid perilous surroundings. This favorable circumstance may account for the fact that in America the appliances for the protection of the laborer have hitherto been given far less attention than in other countries. This paper deals with conditions prevalent in cement factories in Germany and the means by which the Government protects the laborer.

Every employer of skilled and unskilled labor is obliged so to arrange and maintain the workrooms, working contrivances, machines, etc., as to cause the working men or women to be as much protected against any dangers as is feasible. The German government employs different hygienic and commercial committees to inspect at regular intervals all factories and to see that all contrivances are being applied and in good working order. The method is vexatious and very frequently exceeds the rational and practical limits.

Dangers menacing the laborer in a cement factory are due: (*a*) to dynamite explosions, to falling rocks and stones, etc.; (*b*) to accidents occurring in the working of the factory proper, as contact with gears, falling into the crushers and grinding machinery; those connected with the conveyance of the cement from transmission devices or from floor openings; from being burned at the kilns or by ashes and by explosions of coal; (*c*) to injuries from dust, vapors, heat and smoke.

The first does not pertain to the domain of the mechanical engineer and is therefore omitted. Regarding the second, in Germany protecting appliances are included in the supplies furnished by machine factories since the manufacturer is the person logically fitted to know the parts needing protection. The construction of these devices must above all be adapted to render any accident from the working of the respective machine impossible; they should in no way impede the ready and efficient attendance on the machine by the laborer and they should be easily removed in case of repairs, and easily replaced; they should be stable, light and not complicated, and have a pleasing appearance. The paper treats in particular of protecting devices for the following: Jaw crushers, gyratory crushers, reservoirs, gears, belts, rollers, keys, shafting and lubricating system. No loose, torn or waving garments should be worn by the workmen. Many accidents result from faulty distribution of light, narrow and angular corridors, awkward and unhandy arrangement of the transmission contrivances, belts and main starting devices. In cleaning the machinery the dust should be removed automatically. Coal explosions can be obviated by cautious kindling of the kiln and by the application of a feeding contrivance adapted to mix the coal with air in the right proportion. The bin should be built fireproof and be well closed.

The obnoxious effects of the heat are best counteracted by spacious buildings with high windows easily opened and by efficient ventilation. The means of protecting the laborer from dust are many and it is a moral obligation on the part of the owners of cement factories to obviate this cause of physical injury to the fullest possible extent.

METHODS AND APPLIANCES FOR PREVENTION OF ACCIDENTS IN CEMENT PLANTS

By J. G. Bergquist, Buffington, Ind.

Member of the Society

Accidents are caused either by unsafe conditions, or by carelessness on the part of the workmen; hence the remedy resolves itself into (*a*) safeguarding dangerous places, so that any man exercising ordinary care is not required to take any risks in the performance of his duty, and (*b*) promoting carefulness throughout the whole organization.

In dealing with the first, the paper shows by means of numerous views the methods employed by the Universal Portland Cement Company of Buffington, Ind., for safeguarding its distribution of power system, switchboards, lathes, emery wheels, dryers, gears, rolls, shafting, etc.; also the use of overhead walks (containing toeboards), screens, etc., to prevent workmen being hit by falling tools from above, etc.

The danger from the explosion of dust in the atmosphere of the coal-grinding department was removed in this plant by having all elevator casings and conveying apparatus made dust-tight.

As to what is being done to promote carefulness among the men, a five years' record is required from every skilled workman. The newcomer is given one of the company's books of rules on safety, is required to sign a receipt for same, and to pledge himself to live up to it. The book for superintendents and foremen contains 75 pages, on such subjects as the safe load for the various kinds of tools and appliances, like ropes, block and tackle; the book for the other employees contains 25 pages, printed in five different languages.

Constant vigilance is exercised and rewards of merit of various sorts distributed in order to have the rules enforced and to keep up the interest and coöperation of the foremen and the men under them.

DEPRECIATION AND OBSOLESCENCE IN PORTLAND CEMENT PLANTS

By Holger Struckmann, Kansas City, Mo.

Member of the Society

To arrive at a fair percentage to be charged for depreciation, it is necessary to determine the probable average life of the machinery and buildings; this, however, is very difficult as in all cases there is a great deal of variation in the type of machinery and the character of material to be handled; furthermore, the rapid development of the industry has made it necessary to make continuous changes and improvements in the plants and as a matter of fact has caused the depreciation due to obsolescence, to be greater than that due to wear and tear.

The argument is heard quite frequently that if a machine is maintained in a state of efficiency by constant repairs to it, and renewals of parts made, it does not depreciate so long as it does the work and that it is just as valuable when ten or twenty years old as it was when

new. This argument is, however, wrong, as machinery does grow obsolete and cannot be made modern by repairs and renewals.

It is absolutely necessary to create a fund equal to the amount of the depreciation made from year to year, to be kept intact for the purpose of replacing the machinery and buildings at the end of their "life," and this provision should be made without any reference to profits or losses of trade, as depreciation for wear and tear and also obsolescence is a part of the working expense of a business.

The danger of not providing adequately for depreciation and particularly of basing same on the year's trading, is that in bad years the amount which should be set aside for depreciation is paid out in dividends, which means in reality, that dividends are paid out of the capital, a procedure followed as a rule in the expectation of better times when matters can right themselves. Nevertheless this practice is always dangerous and invariably proves disastrous, as might be expected.

A large percentage of the cement mills of this country are equipped today with machinery which is out of date and is a serious handicap, but as provision has been made by scarcely any of the manufacturers to replace it, they must continue to operate old, and in some cases, obsolete machinery. Where the capital stock is raised for this purpose the new machinery quite frequently has to produce sufficient profits to pay interest on the lost capital as well as on the increased capitalization.

It would be far better to distribute less and set aside more for depreciation than to live in a "fool's paradise" and awake to find that the time has come when the machinery must be modernized to meet competition and that the funds to do this are non-existent.

DISCUSSION

In the discussion which followed the reading of the papers, Holger Struckman said, in regard to electric drive: About a year ago we put an ammeter on each individual machine in one of our plants, and instead of correcting the mistakes in the machine through the ammeter, we made the machine come up to the ammeter. In other words, instead of having the variation in load, we kept the load in the machine exactly at a certain figure in accordance with the ammeter, and found that by getting away from the overload we had less belt trouble, and by getting away from underload we had greater output. Mr. Dudley called attention to the fact that in cement mills the surface

of the belt becomes coated with cement dust, and that necessitates considerably higher stresses.

As regards the question of obsolescence, H. S. Spackman stated that if improvements are made at the same rate as during the last ten years, unless the entire cost of the cement machinery, that is machinery used in grinding, etc., is depreciated every five years, and the grinding machinery proper every two years, the average cement mill will certainly spend in changing in that time what it costs to build a new mill.

In discussing the dust problem in cement manufacture, W. H. Mason told of his experiment in that line with a kiln producing about 800 bbl. a day. He constructed a concrete chamber about 60 ft. long, 30 ft. high, and 20 ft. wide, which was really an extension to the kiln. The gases, after leaving the kiln, went all through this chamber, and then up through the stack to the end of it, and a good deal of dust was caught, but since a great deal of dust was still escaping through the stack, a set of sprays was put in the stack, and from these sprays alone, besides what was caught from the other apparatus, about 15 lb. per bbl. was secured, and in some cases even more. There was, however, a continous stream of rain coming down from the kiln on one side or the other, depending on the way the wind was blowing.

No. 1332

THE CORE ROOM: ITS EQUIPMENT AND MANAGEMENT

By Henry M. Lane, Cleveland, O.

Member of the Society

The object of this paper is to bring to the attention of mechanical engineers the fact that by proper study of details in the foundry and core room, it is possible to produce castings better suited for machine construction than those now ordinarily furnished, particularly in regard to their interior or core surfaces.

2 For the present purpose a core may be considered as any body of sand that is formed apart from the mold and then introduced into the mold during its construction or after the mold proper is finished. The function of a core is to form certain faces of the casting, either interior or exterior.

3 Common sense dictates that the faces formed by the core should be as good as those formed by the body of sand comprising the mold proper. This, however, is not generally the case. Until recent times wooden boxes were largely used in making cores and little care was taken. The baked cores were filed to fit one another or the mold. No attempt was made to produce very accurate holes; in fact, the machine shop frequently preferred to cut openings from solid metal rather than to contend with irregular and poorly cored holes full of adhering sand and scale. There is no excuse for practice of this kind and a machine shop has a right to demand as perfect a finish on the interior of a casting as on the exterior.

4 Some of the best core-room practice is found in the specialty shops, such as of manufacturers of radiators, pipe fittings, gas stove burners and automobile engines. In these shops the equipment has been perfected to such an extent that the castings are turned out true to size within limits of a few hundredths or thousandths of an inch, proving that much thought has been spent on the core room and its

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

equipment. But even in the finest and most progressive plants we find conditions which cause loss of time and money. In some plants where the engineers have given painstaking care to the design and construction of the core boxes and core driers, and are using various types of core machines, they are using sands of such a nature that the ingredients of the sand destroy a considerable portion of the oil or other material used as a binder, necessitating an excess of binder, which means unnecessary expense. Most core rooms, also, are hampered by some one condition which is considered a fixture, such as an antiquated oven, a cheap local sand or the prejudice of the foreman in favor of a certain binding material.

5 For the past four or five years the author has given particular attention to the problems of the core room. While many foundry friends have been ready to try experiments at his suggestion, there has been difficulty in harmonizing their results, or in harmonizing results in the same plant when the experiments were carried on a few weeks apart. All have felt the need of improvement in this line, but it required someone to act as a clearing house of ideas, to plan the tests, to see that they were carried out in different plants, and above all, a central laboratory which could unite the results of the different experiments.

6 At this juncture the Robeson Process Company tendered the use of its well equipped chemical and physical laboratories at Covington, Va. and at the author's suggestion set aside a large room for the installation of core ovens and testing machines. With this laboratory available, the coöperation of other manufacturers and dealers was asked and they have all rendered hearty assistance. In another part of this paper will be found a list of the sands furnished and tested and a list of the binders tested.

THE LABORATORY AND EQUIPMENT

7 A general description of the laboratory equipment and methods used will prove helpful to a more ready understanding of the paper.

8 *Sand Mixing.* All of the sand samples are thoroughly mixed before the batches of cores are made. There is no regular mixing mill available and so the following method is adopted: The dry sand is measured out, the binder added, and the mass worked over by hand until the binder seems to be thoroughly distributed throughout the sand. In working by hand the sand is rubbed between the hands,

and the desired amount of water for tempering then added. A careful record of the exact amount of water used should be kept in all cases so that the experiments can be duplicated with exactness. The sand is then taken on an iron plate and mixed by rubbing with an iron pipe as in Fig. 1. It will be noticed that the sand is made into a pile and a little of the end of the pile cut off with the pipe and rubbed forward at each stroke.

9 This method of mixing molding sands was used by Mr. Ronceray in illustrating the advantage of milling sands in a paper which he read before the American Foundrymen's Association at

FIG. 1 MIXING SAND ON A PLATE WITH AN IRON PIPE

the Philadelphia convention some years ago. By this means, a good molding sand was made from clay and sharp sand.

10 It is a rule at the laboratory to mix each batch of sand by cutting it over five times on the plate in addition to the hand mixing. The batch is then taken to the core bench or core machine, as the case may be. Very large batches have to be worked over a little at a time on the plate.

11 *Core-Making Equipment.* A general view of the core-making bench and a portion of the core department is shown in Fig. 2. The central portion of the table is used for making cores by hand. At the ends of the table are mounted two core machines, the one at the right being a Wadsworth screw-feed machine, and the one at the left an Acme plunger-feed machine.

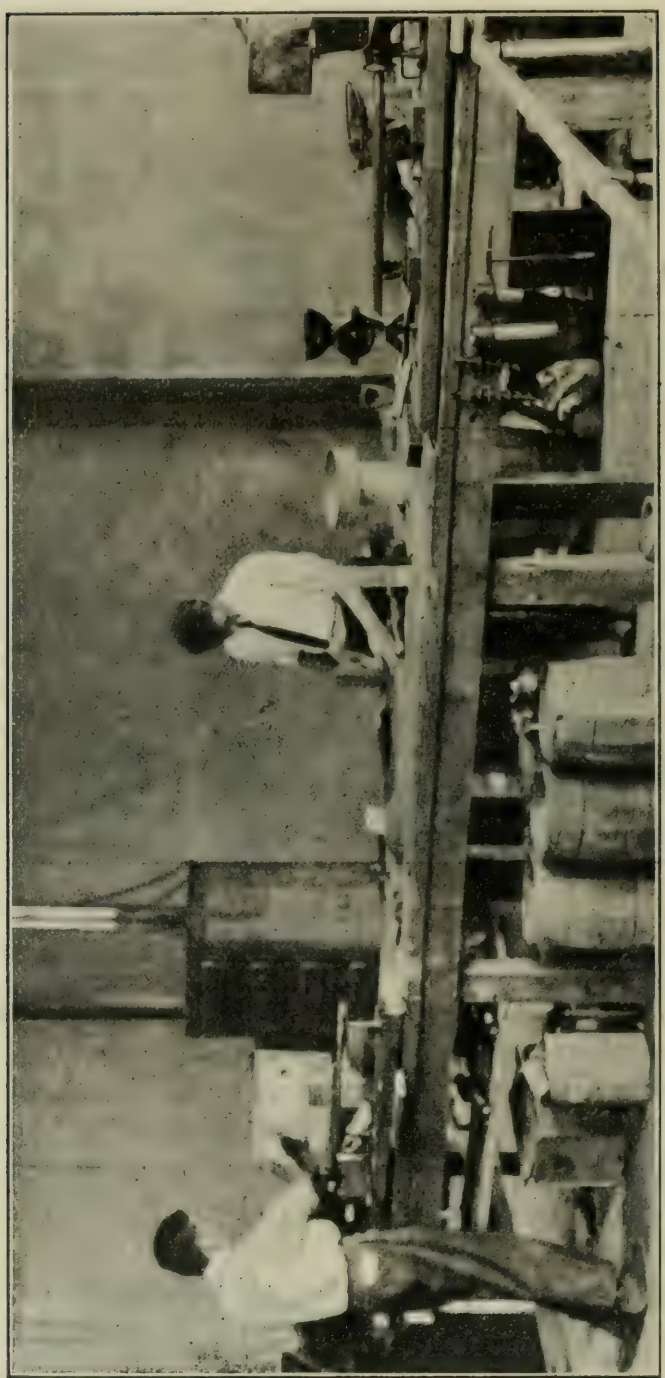


FIG. 2 CORE-MAKING BENCH IN TESTING LABORATORY

12 Both the machines are mounted on heavy planks hinged to the table at one end and arranged to be raised to various elevations

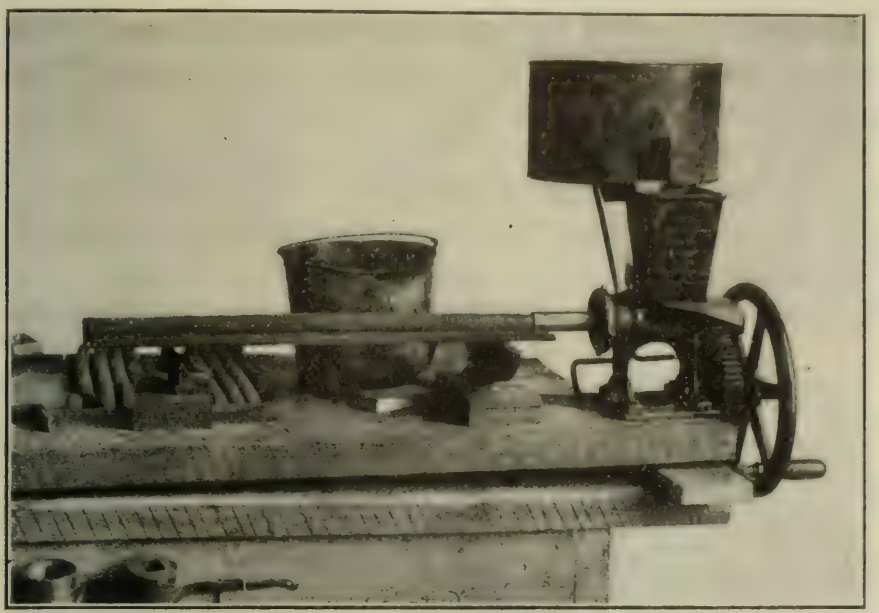


FIG. 3 WADSWORTH SCREW-FEED CORE MACHINE MAKING 1-IN. ROUND CORES

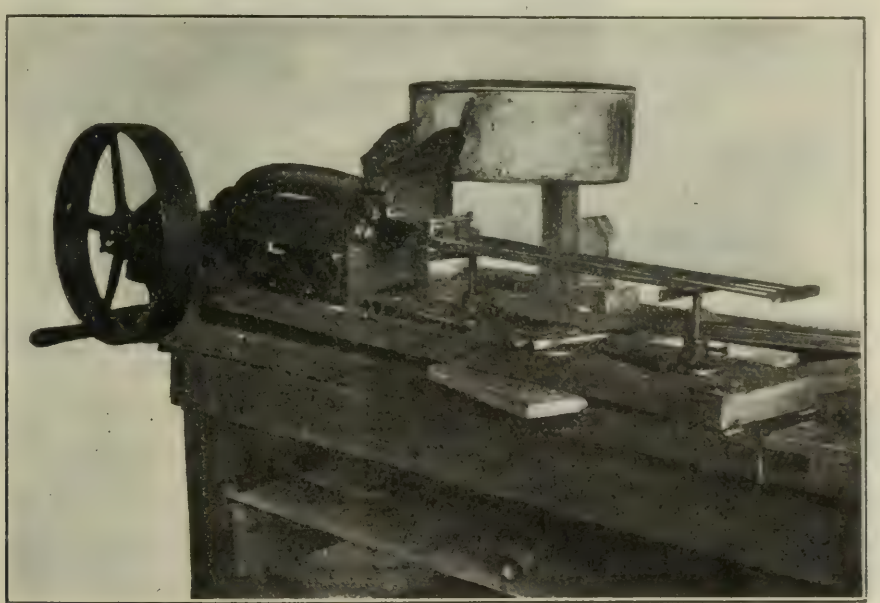


FIG. 4 ACME PLUNGER CORE MACHINE FITTED FOR MAKING 1-IN. SQ. CORES

at the other. This is better shown in Figs. 3 and 4. The device enables the machines to be tested when feeding the cores out on a level surface and also when running the cores down hill at various

angles. This method of mounting was first called to the writer's attention while experiments were being carried on with one of the Acme plunger-feed machines in the making of cores of gangway or old core sand at one of the plants of the International Harvester Company. They found it was necessary to feed the cores down at an angle of 10 deg. to prevent buckling, but by so doing were able to use a very cheap mixture composed entirely of old sand and a relatively small amount of binder. In Fig. 3 the Wadsworth machine is shown making 1-in. round cores and forcing them out on corrugated plates. Fig. 4 shows the Acme machine fitted for making three square cores at a time, and also illustrates the rod

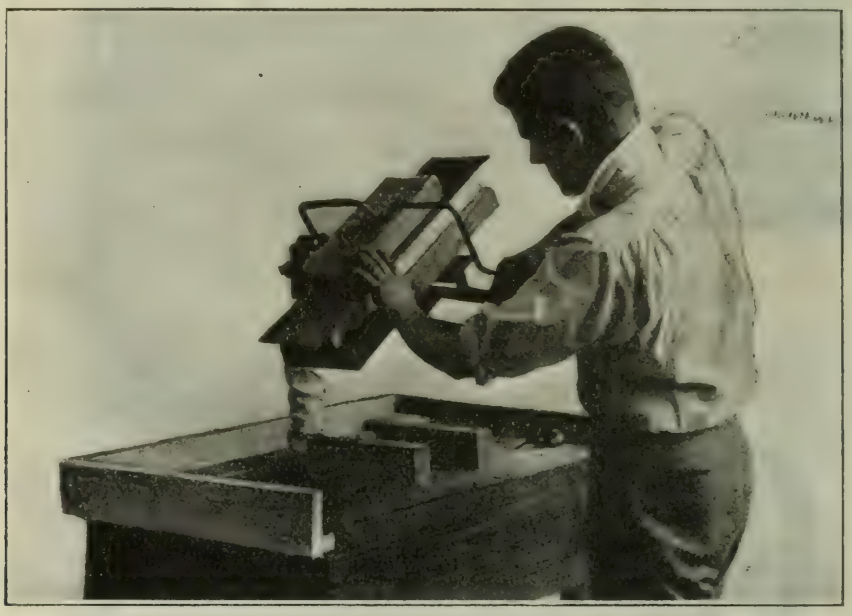


FIG. 5 DETROIT JAR-RAMMING CORE MACHINE IN THE ACT OF ROLLING OVER

guide for keeping the square cores straight after they are run out on the plate; round cores are run out on a corrugated plate. The Wadsworth machine is also fitted with square core dies and in this case the cores are straightened by means of a straight edge. To test the advantage of jar-rammed cores the jar-ramming machine shown in Fig. 5 was installed, which shows it in the act of being rolled over, the core having been formed, the plate placed upon it and clamped. The machine is then raised, rolled over, and let down on to the core plate when the core box is removed and the plate passed to the oven. This machine is used largely for making cores 2 in. square 15 in. long for breaking on supports 12 in. apart. This larger size of core was adopted to test out mixtures used for heavy cores.

13 *Core Testing.* Much thought was given to the proper method of testing cores and several devices have been tried. For testing cores in tension an ordinary concrete testing machine is used with the standard tension pieces made in the ordinary briquette mold shown in Fig. 6. These briquettes are 1 in. thick and 1 in. wide in the narrow part, so that they have an area of 1 sq. in. at the point where they are broken. The cores are pulled in a small testing machine shown in Fig. 7. This machine, however, has a capacity of only 56 lb. and it was found that many of the cores could not be broken with it. It is exceedingly delicate, however, and so is fitted pre-eminently for testing delicate or soft core mixtures such as are used in aluminum foundries or crucible steel work, etc.

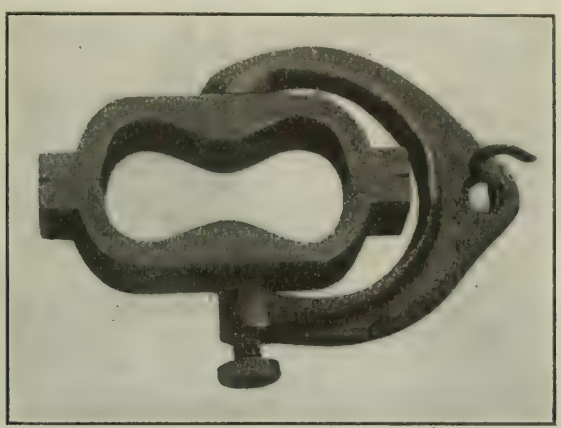


FIG. 6 BOX OR MOLD USED FOR MAKING TEST CORES FOR TENSION TEST

14 For testing cores which develop greater strength the device shown in Fig. 8 was first constructed. This consists of two supports, the tops of which are formed with a $\frac{1}{4}$-in. radius, giving a $\frac{1}{2}$-in. circle, the bearing points of the supports being exactly 12 in. apart. A 1-in. square core 15 in. long is made in the core box shown at the bottom of Fig. 8, and the core then laid across the top of the supports with the shackle in the center as shown. The shackle is located by means of the piece of wood shown at the right which has a notch cut in it so that when the shackle bears against the end of the stick the $\frac{1}{2}$-in. bolt will be exactly in the center of the core being tested. A pail is hung from the shackle and shot poured into it until the core breaks. The shackle is then dropped into the pail and the pail, shot, and shackle weighed on a spring balance, thus giving the weight which it took to break the core.

15. It was found that considerable care had to be taken to conduct the testing at about the same rate, that is, to pour the shot at about the same speed, so that the weight would come on to the core uniformly and at a given rate. Where the shot was poured very slowly the core would break at a much lower figure than where the shot was poured rapidly.

16. The use of a Fairbanks cement testing machine was secured at the Industrial Testing Laboratory of Cleveland and in Buffalo at the

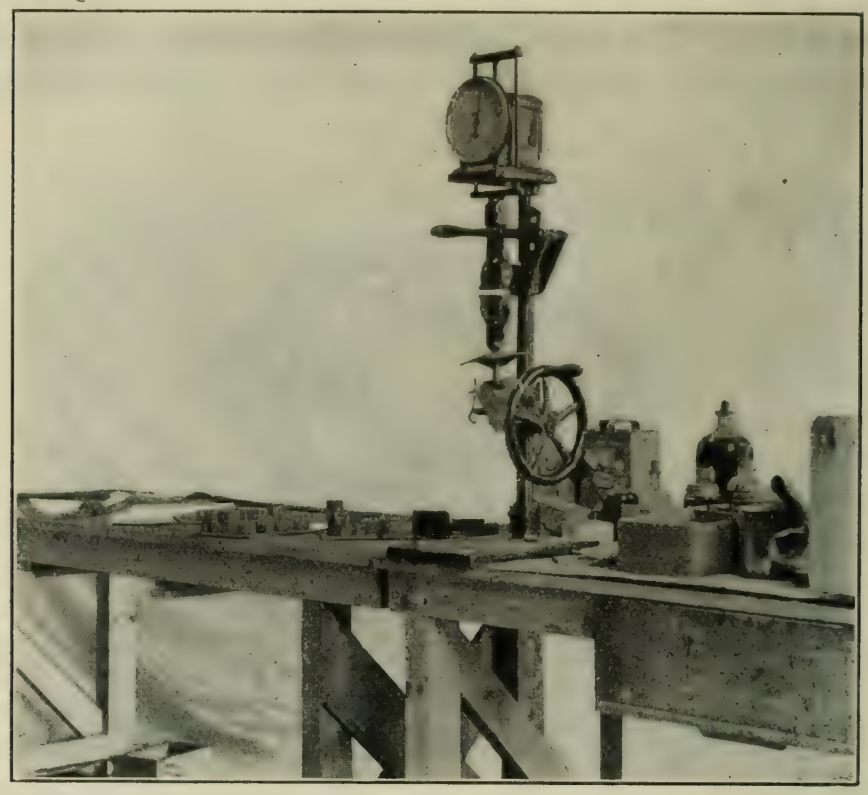


FIG. 7 TENSION-TESTING MACHINE USED IN CORE EXPERIMENTS

Buffalo Testing Laboratory. Series of cores were broken on these machines to test commercial mixtures.

17. The author believes the tension test to be the fairest comparative one for a core as it can be applied more uniformly and easily than a transverse test. Cores are also more delicate when tested in tension, and develop greater strength, and small errors do not make as great a relative difference. Core mixtures were found in use in foundries the tension strengths of which varied from less than 3 lb. per sq. in. to 150 lb. per sq. in., and apparently the necessity of meeting the varying needs of castings thoroughly justifies this wide range in strength.

18 *Baking Equipment.* For baking the cores a small gas-fired oven was first used, which could be kept at a temperature of 400 to 410 deg. without difficulty. When larger cores were made, however, and tests of mixtures compared, it became necessary to bake batches of considerable size at a time, and so a Wadsworth portable core oven was installed, as shown in Fig. 9. This illustration also shows the Bristol recording thermometers attached to the core oven. One of the thermometer bulbs was placed under the top shelf and the other at the top of the bottom shelf. Iron pipes of $\frac{3}{4}$ in. size were carried through the back of the oven so that the thermometer tubes were at about the center of the oven.

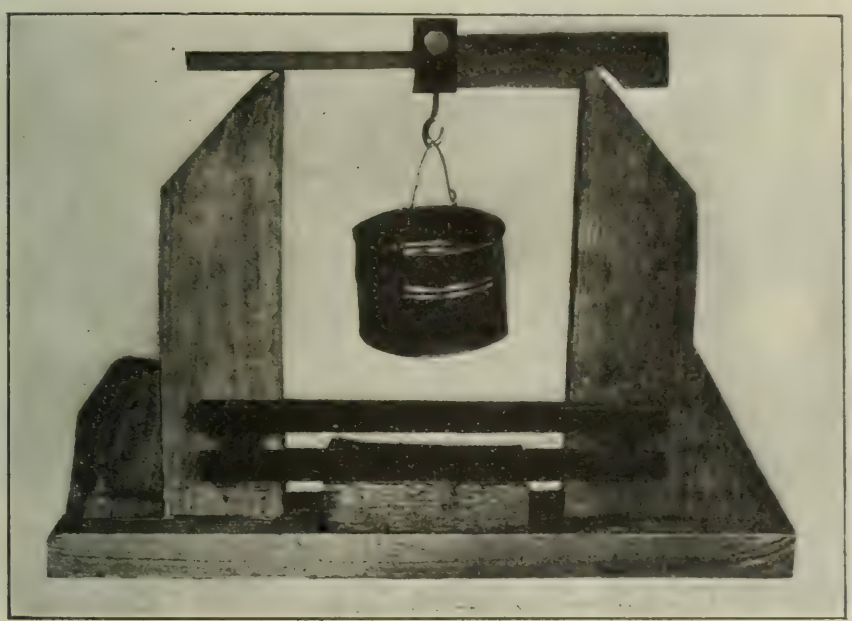


FIG. 8 DEVICE FOR BREAKING 1-IN. SQ. CORES

19 At first there was trouble in getting the required temperature in the oven, but by burning anthracite coal no further difficulty was experienced in maintaining the temperature at 400 to 410 deg. at either thermometer. In ordinary practice, however, when the vent at the top of the oven is wide open there is a difference of from 30 to 40 deg. between the two thermometers in this oven, the lower thermometer being the hottest. This is due to the fact that the cores on the upper shelves absorb heat and that the lower shelf is directly over the firebox. The lower thermometer is therefore affected at times by some directly radiated heat, and this would also affect any cores being dried, as the radiated heat would strike the bottom of the core plate and tend to dry cores from the bottom up.

20 The Bristol thermometers attached to the core oven in Fig. 9 have also been used in testing the temperature in other ovens in different foundry plants.

21 *Fineness Tests.* To determine the relative proportion of the different sized sand grains in a given core sand it was necessary to conduct fineness tests. This was done with a set of Tyler standard sieves, of 20, 60, 80, and 100 mesh. A fineness test was made on all

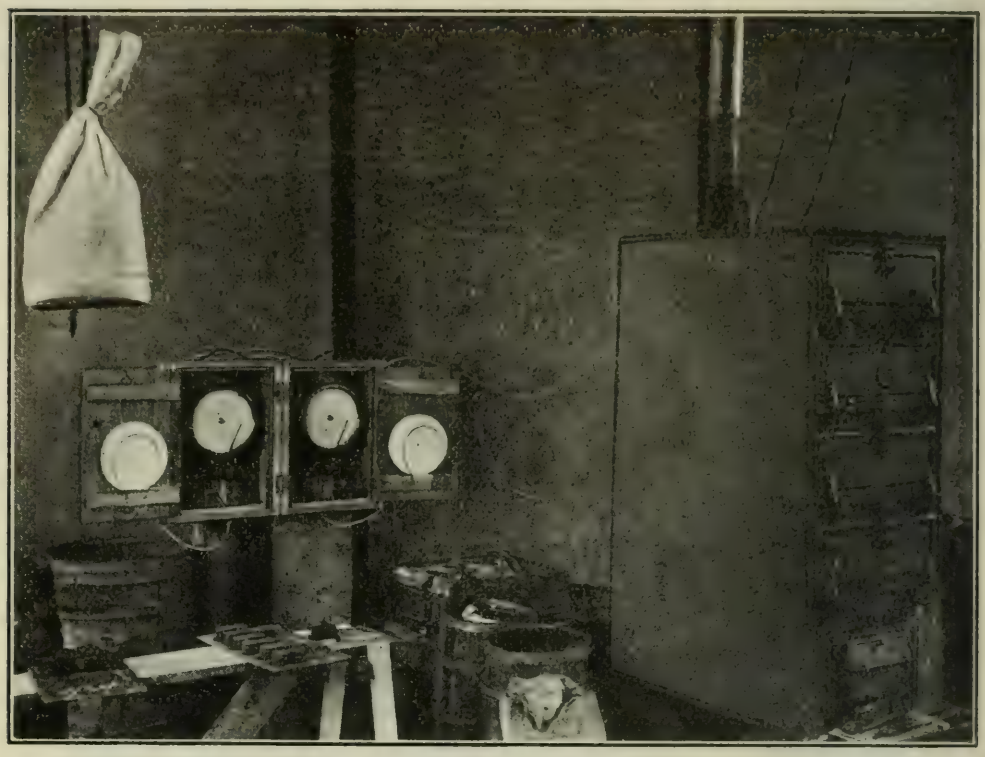


FIG. 9 WADSWORTH PORTABLE CORE OVEN WITH BRISTOL RECORDING THERMOMETERS ATTACHED

sands tested and was found to have an important bearing on the results that could be obtained with any given sand.

22 Other tests than those conducted at the Covington laboratory, as well as microscopic and photographic work and other investigations have been made possible by the assistance of various firms and individuals. Several of these to whom the author is especially indebted, together with firms who supplied apparatus for the laboratory, are mentioned at the end of the paper and to them the author expresses his thanks and appreciation.

ACTION ON A CORE IN THE MOLD

23 *Introduction of Metal into a Mold.* Before proceeding with a discussion of the subject it may be well to state what happens when a core is surrounded with metal. The action is shown in Fig. 10, which represents a portion of a cylindrical casting in the process of pouring, at a point where the mold is about two-thirds full, the metal rising over the core. Metal enters the gate at the left and flows to the bottom of the opening under the core.

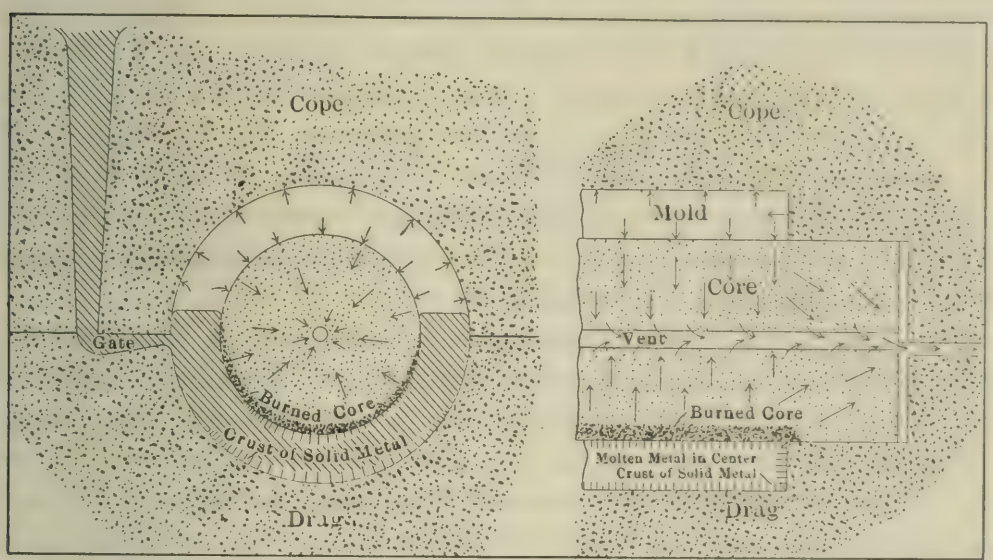


FIG. 10 MANNER IN WHICH A CORE IS ACTED UPON BY THE METAL AS IT ENTERS THE MOLD

24 At first the mold is full of air and the moment molten metal enters, this air becomes highly heated and expanded, thus instantly creating a pressure in the mold. The inflowing metal must not only expel a volume of air equal to the volume of the mold, but the expansion of the air greatly increases its volume. Then also, considerable steam is generated as the metal enters, and this too has to be expelled. This mass of air and steam escapes through both the mold and the core as indicated by the arrows in the upper portion of the illustration. As the metal fills the mold the gas pressure on the inside is relieved, but a new set of conditions appears which demands a porous mold and core.

25 If the metal is poured at the proper temperature, a skin or crust of solid metal forms almost instantly on the sides of the mold and the core (see Fig. 10), which prevents the passage of air or gas from the core or mold into the metal, or from the metal into the sand.

At first, however, the skin is rather flexible, and should gas be generated in either the mold or the core more rapidly than the vent can take care of it, it may blow back and scab the casting.

26 As the metal rises against the bottom of the core its surface is burned. As a core is dry, no steam is generated during this operation, but the hydrocarbon compounds in the core binder are driven off and forced through the core toward the central vent, as indicated by the arrows in Fig. 10.

27 *Gases Formed in Cores.* As long as air is present in the mold or core all forms of carbon are burned or oxidized so that the first gases to be expelled may contain carbon dioxide. As it becomes entirely surrounded with metal the air is quickly exhausted and from that time on only such hydrocarbon gases as can be distilled off by heat are expelled. From this it will readily be seen that the volume of gas generated in the pouring of metal around any core is largely determined by the volume of volatile hydrocarbons which can be distilled from the core binder.

28 The baking temperature to which the core is subjected will have an important bearing upon this subject. Certain binders can be baked at a high temperature so as to expel most of the volatile hydrocarbons and leave little but solid carbon as the core binder, but where this is done, an excess of binding material must be used, as the binding power of the hydrocarbon compounds is largely sacrificed. Then too, if the baking is continued until all of the volatile hydrocarbons are driven out, the volume of the binder is changed to such an extent that cracks or checks will generally be formed in the core. The ideal binder is one which contains sufficient hydrocarbons to give a strong core, but it must not give off during pouring gases which are injurious or even trying to the workmen. This subject will be referred to later under binders.

29 *Refractory Base of a Core.* The refractory material of which the core is composed should be of such a nature that it conducts heat but slowly, so that the rate at which the succeeding layers of hydrocarbon are distilled off is sufficiently slow to enable the vent to take care of the gases as they are formed.

30 There are some cases in which it is advisable to increase the conductivity of a core so as to cause it to act as a chill on the metal and when this is desired the binder problem is still further complicated.

31 After the metal has come in contact with the core and commenced to burn out the hydrocarbon compounds, there should still be some binding properties left to act until the skin of metal next

the core becomes sufficiently strong to resist any tendency of the molten mass of metal to cut the core.

32. Where metal must flow through or cover a core it is difficult to prevent cutting or washing by using bonds of a carbonaceous nature only. Clay or a refractory bond of similar nature has to be resorted to; sometimes the surface of the core is protected by silica wash or clay and blacking wash. If the binder is present in the core in sufficient quantity to leave a fairly large percentage of fixed or free carbon, the metal is not liable seriously to eat into or burn into the core. In many cases this condition necessitates the use of higher percentages of binder than would be required simply to furnish the necessary strength in the core.

DIVISIONS OF THE SUBJECT

33. To avoid complication the subject is divided into sections and each treated separately. The sections selected are as follows: core-room location and arrangement; core sands and core binders; selection and compounding of core materials; core ovens and core drying; core pasting, core handling and core storage; core machines and core-room rigging.

CORE ROOM LOCATION AND ARRANGEMENT

34. The location and area of the core room are of more importance in the production of castings than most foundrymen realize, since the modern tendency in certain classes of work is to throw an ever increasing responsibility on it, and some complicated castings such as air-cooled automobile cylinders are made in molds that are composed entirely of cores. The definition of a core as already given was carefully chosen with this phase of the subject in mind. The location depends upon several factors: the size of the individual cores, the number of pounds of cores used per molder per day, the strength of the individual cores and the method of handling the cores from the core room to the molders.

35. *Core-Room Area.* A series of observations of the proper area of the core room, extending over about fifteen years, has shown that the core room varies in area from 10 to over 50 per cent of the molding floor. This includes the space occupied by the ovens and core storage but not the core sand storage. In very heavy grey iron work, such as large Corliss engine cylinders, heavy machine-tool castings, large forging machines, etc., the space devoted to core making may

be equal to at least 60 per cent of the active molding floor space, but in this kind of heavy work about half is usually inactive, because heavy castings must be left in the sand several days to cool, hence the area as compared with the molding floor is about 30 per cent of the total.

36 The core-room proportion in the case of side floorwork in a grey iron foundry is generally much less because many of the molds require no cores and as a rule the floors are cleaned each day, so that the entire molding area is continually active and varies from 10 to 15 per cent of the area of the side floor.

37 In the case of loam work frequently only a very small number of cores are used which are not made on the loam floors, but of course when loam work is being carried on drying ovens are necessary and the entire proposition partakes of the nature of a core problem.

38 For light grey iron or malleable work the area of the core room depends entirely upon the character of the product. The fact that in the core storage the cores are kept on shelves one above another and the small ones are dried on cars having several shelves, tends to reduce very greatly the total area required. The cores for a given mold usually take up much less space than the mold itself and they are not accompanied by flasks. For grey iron work on automobile castings, including cylinders, the core department will have an area varying from 40 to 60 per cent of that of the foundry. In aluminum work for automobiles it is about 40 per cent, in brass jobbing shops 10 to 15 per cent, for brass fittings 20 to 25 per cent, while for iron fittings it takes from 20 to 45 per cent.

39 The core-storage space to be provided is also important. In the case of jobbing work comparatively little room is necessary, since the cores when completed are usually sent to the molders' floors. Where a line of standard work is being manufactured, provision should be made for carrying at least one-half of a day's supply of cores, and where night ovens are used for baking an entire day's supply.

40 *Handling Warm Cores.* In order to avoid rehandling whenever possible, cores should be sent from the core-oven trucks to the molders without placing them on storage shelves as this involves an extra handling. This question, however, brings up the advisability of handling cores hot. With many binders under such conditions more binder will have to be used than if they are allowed to cool.

41 One large foundry in this country using a great many oil-sand cores changes its oil-sand mixtures at about 3 p.m. by decreasing the amount of binder 15 per cent. The reason for this is that

the cores made subsequently will be run out of the core oven on to the plates and left to cool over night before they are handled, so that the full strength of the binder will be developed. During the day they are ordinarily taken from the plates hot and carried to the molders. In many cases a saving of 15 per cent of binder would more than pay for the rehandling of the cores and the placing of a core storage between the core department and the foundry.

42 *Transportation of Cores.* Where the number of cores used per molder is so great that the weight of the cores delivered to each approximates the number of pounds of castings turned out by him per day, the problem of transporting the cores becomes serious and must be taken into consideration in locating the core room. In this there must also be considered the element of breakage. Every core that is broken represents not only the loss of the sand and binder but also of the time expended in making it, the fuel for baking, and other labor in the way of transportation. Every possible precaution should therefore be taken to reduce this item.

43 If the cores are large and delicate, such as those required in aluminum or light crucible steel castings, they must give way readily before the shrinking metal and should be handled as little as possible, hence the core room should be near the molders.

44 If the cores are standard and sufficiently hard so that they are handled in boxes or in piles on boards as in some fitting shops, they will stand transportation a long way, and here it will pay to centralize the core department, equip it with labor-saving machinery, and reduce the cost to a minimum. The cores can then be distributed to the various departments without fear of serious loss through breakage. In the case of heavy work where they have to be handled with a crane, if there are several separate bays or departments it is frequently more economical to provide each section of the foundry with its own core room and to arrange it so that the main traveling cranes handle them from the core oven trucks to the molds at one operation.

45 For transporting cores about the foundry we have platforms commonly called boats, which are rectangular in shape, made of plank, with heavy battens on the back, supported at four corners by slings from the crane hook. The cores to be moved are piled on the platform and shifted to the parts of the foundry where they are to be used. For handling and transporting medium or fairly heavy cores a device of this kind saves rehandling and guards against breakage. It also does away with confusion in the gangway due in many plants to carrying of cores to the various floors. In some foundries the cores

are carried to molders on a trolley system in boxes or specially prepared cages supported by spiral springs. In other cases spring trucks running on the gangway floor or spring-supported platform cars running on industrial railways are used for the purpose.

46 In the case of medium heavy work, cores weighing 100 to 200 lb., the core-oven truck may be arranged so that each shelf can be lifted off by the traveling crane and used as a boat to carry the cores to the floor; in other cases the entire core car may be picked up by the cranes and delivered to the molders floor.

47 *Handling Core Sand.* As the core sand is bulky the problem of handling it assumes considerable prominence in most plants, and this necessitates the location of the core room in such a position that the sand can be delivered from the railroad cars into the storage bins with the least amount of handling. For the most effective work the sand should be as dry as possible, hence it should be put into storage at a dry season of the year, and at least a nine months' supply provided for. Buying the sand at this time of the year, also reduces freight costs by eliminating water contained in the sand during rainy periods.

48 To sum up the matter, the location of the core room should be such as to minimize the handling of the cores, being nearer the molders in the case of very heavy work handled by the main cranes, or in cases where delicate cores are used, as in aluminum or light crucible steel work, and being centralized when they are strong enough to stand transportation.

49 The area will depend upon the product, and a fair estimate may be obtained by taking the number of cores required per man per day, and estimating the number of cores the coremaker will make per day, thus arriving at the number of benches required. The space occupied by coremakers' benches is as a rule about 35 to 40 per cent of the total core-room space, the balance being taken up with ovens and storage racks for green and dry cores.

CORE SANDS AND CORE BINDERS

50 The term core sand covers a wide variety of materials adapted to different classes of core work, and the conditions to be met in the core room necessitate the use of a number of grades of sand in most core rooms. In all core sands the principal heat resisting element is silica, the silica grains also forming what may be termed the backbone of the core. Ordinary molding sand rarely contains over 80 per cent silica, the balance being made up of alumina, oxide of iron, and other

impurities. In a molding sand a certain percentage of clay is necessary to form the bond. They may be divided into bondless or sharp sands for oil-sand mixtures or cores which require a great deal of vent; and the bonded sands and gravels. The sharp sands contain from 97 to 98.5 per cent silica, while the bonded sands contain from 97 to less than 90 per cent.

51 *Proper Condition of Bond in a Core.* Since a core must be sufficiently porous to vent freely, the sand composing it must be of such a nature that there will be a considerable percentage of voids or open spaces through which the vent may escape. In any core the individual grains of sands must be bound one to another with some material, collected, if possible, at the contact points thus leaving the spaces between free for vent passages. Anything that tends to roughen the surface of the grains of sand in the passages tends to retard the vent.

52 *Form of Sand Grains.* In concrete and masonry work sharp sand is desirable on account of the fact that it affords a better grip for the cement or lime in the mortar. For core work rounded grains of sand are the best because they give a maximum of vent passages and larger areas at the contact points for the bonding material to act. The strongest cores that the writer has seen used in practice were made from oil-sand mixtures with thoroughly rounded grains of sand, the individual grains being of approximately uniform size.

53 In concrete work the object is to fill all the voids between the particles of the material with finer stock, so that the final product will be a uniform solid mass. In making cores the requirement is radically different, as it is desirable to have the greatest possible number of fine voids; only they should be so fine that the metal will not tend to follow or flow into them. This necessitates the use of different sized grains in the core sand for different metals.

54 Some of the brass and bronze mixtures are particularly searching and will force their way into cores where iron would lie smoothly on the surface. There are two reasons for this, one that the melting temperature of the brass and bronze alloys is lower than that of the iron mixtures and hence they remain fluid for a longer time in contact with a core; and the other that the iron alloys seem to be more viscous when fluid and will not flow into as sharp corners as the brass and bronze alloys.

55 Aluminum behaves like iron in this particular. Sand mixtures which would be perfectly satisfactory for iron or steel or for aluminum may be wholly unsuited for brass or bronze. A brass or

bronze containing considerable phosphorus is particularly searching in its action, also an iron containing considerable phosphorus is more fluid than other mixtures and has more of a tendency to cut into a core. From this it will be seen that the selection of a particular grade of core sand must depend to a considerable extent on the metal to be cast.

56 *Early Core Practice.* In the earliest foundry practice cores were made from mixtures containing no artificial bonds. A sand was chosen containing enough clay to hold the grains together when dry, and if this showed a tendency to stick to the casting some sawdust or other carbonaceous material was added to the mixture, causing it to fall to pieces next to the casting more easily and clean out better. It is probable that the first artificial core binders used were rye and white flour and pea meal, and there are also records of the use of sour beer. In those days all molds were thoroughly dried before the metal was poured into them, so that the mold partook of the nature of a core. To harden and strengthen the face of the mold or core, it was frequently sprinkled with sour beer or molasses water and then dried. When the metal came in contact with the surface, it burned out the carbon of the binding material used at the surface and made it easy to clean out the core.

57 *American and Foreign Practice.* In America, demand for a big output and cheap castings has developed a number of lines of foundry practice to an extent not found in any other country. The green sand mold is more universally used here and the variety of core binders exceeds that employed in any other country. At present probably the most common core binder used abroad is what is known as core gum, which is really dextrine, one of the products of the starch industry. Pea meal is also used abroad to a considerable extent and rye flour and linseed oil in certain foreign countries. America has been the first to introduce an extensive line of specially prepared binders.

58 *Kinds of Binders.* There are two classes of binders as regards their action on the sand. These are the true pastes, which do not flow to the contact points of the sand, and the binders that flow to the contact points as the drying and baking take place. All binders which act as a paste are unaffected by clay in the sand. This is also true of resin and pitch. Binders that partake of the nature of an oil are injured and in some cases ruined by the presence of clay in the sand. The advantages of sharp sands and sands of uniform sized grains were never fully appreciated until the complicated problems of the water-cooled automobile engine cylinders required solution. At

first intricate systems of mechanically formed vents were used, but later it was discovered that by using clear silica sands and oil as a binder that the vent would find ample passage in the spaces between the rounded grains.

59 *Fitting the Binder to the Sand.* In selecting the core sand for any given location the problem to be solved is the finding of the cheapest finished core. Cores for a given class of work must have a given strength per square inch and must disintegrate when the metal is poured about them, so as to give ample opportunity for the metal to shrink. Generally in producing a core of this kind a number of courses are open. A local bonded sand may be used with resin, flour, dextrine, black compound, glutrin, or molasses. As a rule black compound or pitch cores are more suited for heavy work than for light. For small work greater strength per square inch of core is generally required than for large work, and for great strength associated with free venting there must be a binder that will hold a sharp sand and at the same time permit the free circulation of the escaping gases. Flour or dextrine tend to a large extent to stop the vent, as do also the black compounds, which contain dextrine as a green binder. This calls for the use of an oil mixture, an oil and glutrin mixture, a glutrin and clay wash mixture, or molasses.

60 A number of different water soluble by-products from commercial processes have been or are used to some extent as core binders. In some places in the vicinity of rum distilleries a product known as distillery returns, distillery slop, or sour beer, is used, in other locations sour beer from breweries; but both are employed only in limited localities as they are weak adhesives and hence not good binders.

61 The principal objection to the use of molasses in the core room is the fact that this binder is not uniform in its adhesive properties. First fermentation has to be dealt with. To test its effect the writer measured out a quantity of molasses, diluted it with twice its volume of water, and let it stand in a crock. The specific gravity of the solution was taken every few days and batches of cores made from it were tested for strength. These fell off rapidly in strength as the fermentation proceeded, losing more than half, and at the same time the specific gravity constantly decreased. Foundrymen have been known to purchase fermented molasses thinking that they were getting a cheap bond. In a case of this kind, however, one has no knowledge of the actual bonding power.

62 At the time the fermentation experiment was made on molasses a sample of glutrin was measured out, diluted to the same proportions,

and allowed to stand in a crock next to the molasses. Cores were made from it each time that cores were made from molasses. Those from glutrin gradually increased in strength with the age of the mixture, due to its slow concentration by evaporation. The molasses mixture was of course evaporated just as fast, but the composition was changing so rapidly from fermentation that the effect of evaporation did not show.

63 Another objection to molasses is the fact that no matter how honest the dealer or how free from fermentation the stock may be at the time it is used, its bonding power varies with the source from which it was derived. In making cane sugar the plant is topped in the field and the tops are thrown away because the juices in the upper part of the stalk have not been converted into sugar. If the stalks are topped too high, a large proportion of the juice carrying no sugar enters the fluid and remains in the molasses after the sugar has been crystallized.

64 As molasses is the residuum of the sugar process it must contain all its impurities, and these vary with the source of material from which the sugar was made, the method of work, the way in which the cane was topped, and many other factors.

65 The other water soluble bond extensively used in the foundry is glutrin, a by-product of paper manufacture by the sulphite process. The sap stored in the cells of the spruce wood used in paper making is extracted by boiling with the sulphite solution and when treated, to remove certain undesirable elements, and concentrated, it becomes the binder known as glutrin. While its composition is complex, consisting of tannins, wood sugars and resin in soluble form, it is a product uniform in composition and hence in binding power, and it will not ferment.

66 *Use of the Microscope.* It has been found that a microscopic examination of the fracture of a core would tell much as to its venting properties and the efficiency of the bond, but thus far, however, it has proved impossible to produce a microphotograph that would show all that the human eye could see in examination of a core, because the focal depth of the micrographic lenses is exceedingly shallow, so that when examining cores at magnifications of 60 or more diameters it is necessary to rack the objective back and forth and examine the top, the middle and the bottom of a grain of sand, thus giving a clear mental picture of the bonding conditions. A microphotograph taken in the ordinary way does not show this as it has not sufficient depth of focus.

67 *Action of Oil in Binding Sands.* Oil of a proper grade is undoubtedly the strongest and, weight for weight, the most efficient binder known when dealing with clean silica grains. In the paint trade linseed oil is considered the best drying oil, and it is also the best core oil. The action of linseed oil in bonding a core is as follows: The material must be so thoroughly mixed with the sand that every grain is uniformly covered with oil or an emulsion of oil and water. When the heat of the core oven acts on the sand the moisture is evaporated and driven off. As the moisture passes through the core toward the outside the oil remains behind on account of its relatively high viscosity and first uniformly coats each grain of sand. By capillary attraction excess of oil tends to accumulate at the contact points of the sand grains and finally dries down here and forms an area of bond somewhat larger than the contact areas. The heavier the oil, that is the more body it contains, the more will the space about the contact points be filleted and the bonding area increased. Most of the other drying oils contain less body than linseed, and hence do not give so firm a bonding mass at the contact points as is the case with linseed. All oil-sand mixtures must be tempered with water.

68 *Action of Paste in Binding Sand.* The radical difference between the bonding of sharp sand with an oil and with flour or dextrine is that the latter forms small masses or grains of paste which dry on the face of the sand grains. These do not flow over it to the contact points, but dry in the place where they were left by the mixing machine. For this reason only those masses situated at the contact points become efficient as a binder. A microscopic examination of a flour or dextrine-bound core shows that certainly not less than 60 per cent of the material is inactive as a binder and is located in such a way that it tends to block the vent passages by giving the grains of sand a rough instead of the smooth varnished surface given to the grains by oil.

69 *Action of Resin and Pitch in Binding Sand.* Both resin and pitch bond sand by melting and flowing over or between the grains, collecting to a certain extent at their contact points as the core cools, but they are not so efficient as is oil. Resin and pitch do not enter into combination with clay, but their binding power is added to whatever power the clay may have. When such a core comes in contact with the molten metal in the mold, the carbon material of the resin or pitch is burned out, which disintegrates the core, thus making it possible to clean it from the casting; while if clay or natural bond alone were depended upon the bond would be hardened instead of softened, and it would be impossible to clean it from the casting.

70 *Action of Molasses in Binding Sand.* The action of molasses in a core depends upon the rate of drying and the temperature. Under ordinary core-oven conditions where cores are put into a hot oven, the water is quickly expelled and the molasses brought to the boiling temperature. When it reaches the consistency of ordinary molasses candy, it boils up in a similar manner and then hardens in thin plates connecting the sand grains into a more or less intricate system, and with the carbon exposed in exceedingly thin surfaces, though of course some of it dries at the contact points. If the core is taken from the oven at the critical point just as these plates are in their strongest condition, it will be found to be strong and serviceable; but if taken out too soon, it will not have developed sufficient strength to give a strong core, and if left in the core oven a little too long the carbon will have become rapidly oxidized and the strength of the core will fall off. For these reasons a molasses core is exceedingly sensitive and with ordinary existing core-oven conditions it is practically impossible to produce uniform results.

71 *Action of Glutrin in Binding Core Sand.* The compounds contained in glutrin are of such a nature that they do not enter into and combine with clay as does oil, but they do form with it an exceedingly efficient emulsion and tend to carry it from the faces of the sand to the contact points. In drying, glutrin behaves more like oil in that it tends to flow to the contact points.

72 In sharp sand without any clay bond the fact that glutrin lacks the viscosity of linseed oil, gives it a tendency to follow the moisture to the surface of the core. In a mixture of clear sharp sand a glutrin core will tend to have a very hard skin and a soft interior. This is well illustrated in the microphotograph shown in Fig. 11, which is a core made with clear Rockaway Beach sand and glutrin in the ratio of 1 part of glutrin to 50 parts of sand. The glutrin tended to sweat to the surface of the core, giving a hard surface as shown by the darker material along its face. The main body of the sand was bound fairly well but this dense layer on the outer surface interferes with the escape of the steam from the interior toward the latter part of the drying operations and may result in an imperfectly dried core. Attention is called, however, to the exceedingly open character of the core itself.

73 This sweating tendency is entirely overcome by introducing from $\frac{1}{2}$ to 1 per cent of clay into the mixture, the clay and the glutrin together forming a compound which is sufficiently viscous so that it draws to the contact points rather than following the moisture to

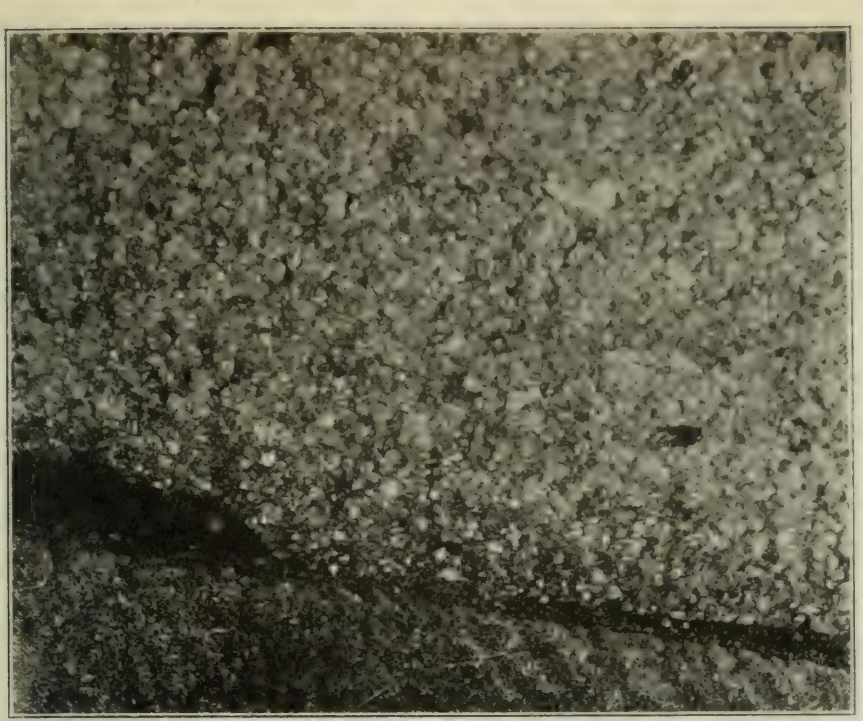


FIG. 11 ROCKAWAY BEACH CORE COMPOUNDED WITH GLUTRIN SHOWING SWEATING OF THE BINDER TO THE SURFACE OF THE CORE

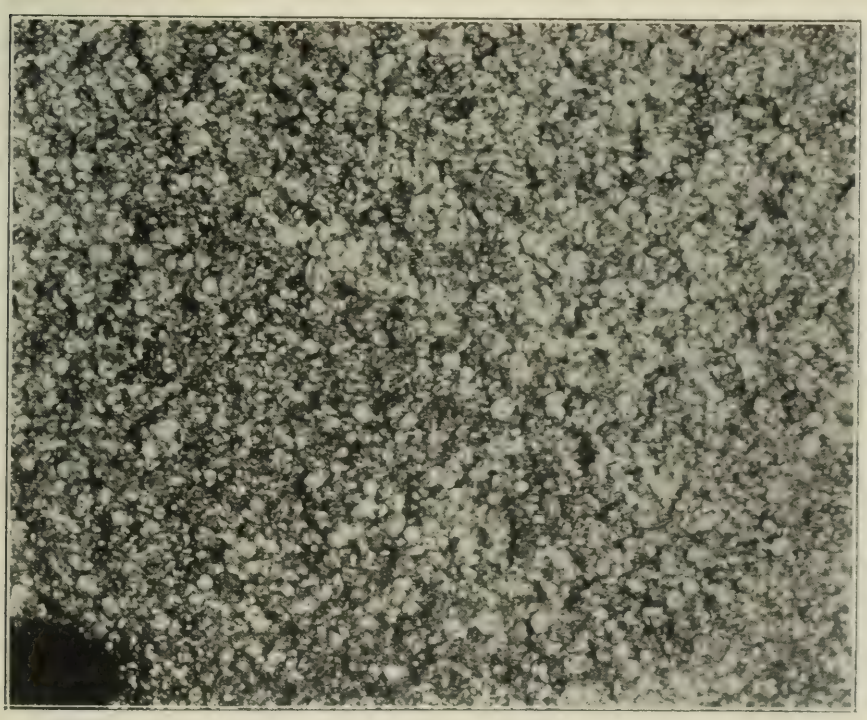


FIG. 12 OIL-SAND CORE FOR GAS STOVE BURNER

the surface of the core, making it strong and at the same time one which does not have the vent passages stopped. Glutrin can also be used in connection with oil in clear sand mixtures, as the oil prevents it from sweating to the surface of the core and the combination blends perfectly to form an efficient bond at the contract points.

74. Fig. 12 shows a microphotograph of a section of an oil-sand core intended for gas stove fittings, which was made from a mixture of 22 parts silica sand, three parts bank sand, and a commercial or blended oil at the ratio of 1 part of oil to 25 parts of sand. The exceedingly open character of the core is clearly shown, but its strength is about 75 lb. per sq. in. in tension. A core of this kind shows absolutely no tendency for the binder to sweat to the surface and as the sand from which the core is made is fairly uniform in size there is a maximum of voids or open spaces for vent passages.

75. Fig. 13 is a pitch bound core intended for heavy work. The mixture is known as a loam mixture, containing Zanesville loam as a base, the proportion of the ingredients being as follows: 12 parts new Zanesville loam, 14 parts old core sand, 4 parts pine sawdust, and 1 part ground pitch. This firm grinds its own pitch and mixes no dextrine with it, grinding it fairly coarse, so that occasionally fairly large spots show in the fracture of the core. It is mixed in a Standard Sand and Machine Company's batch mixer with about 11 per cent of water by weight. When rammed into the molds it has some green binding power due to the excessive amount of clay in the Zanesville loam. The cores are dried out in the neighborhood of 350 to 400 deg. in large ovens which are fired every night. The shrinkage of the sawdust and the driving out of the large amount of water gives the core considerable vent. The pitch and loam form the binders and as the pitch is rather coarsely ground some of it becomes active when the old core sand is introduced into the mixture. Probably at least 50 per cent of the pitch in the old core sand still possesses binding power. A core mixture of this kind would only show a strength of 10 or 12 lb. per sq. in. and in some cases not over 6 or 7 lb., so that the cores have to be strengthened by the use of rods and the face has to be protected by a blacking mixture.

76. Another core of very close composition is illustrated in Fig. 14, a section of a machine-made core intended for use in a brass foundry. The mixture consists of 32 parts of Del Ray bank sand, 32 parts of Lake sand from near Buffalo, 8 parts flour, and 1 part of oil, and is tempered with about 7 per cent of water by weight. The shrinkage of the flour paste together with the expelling of the water

FIG 13 PITCH-BOUND CORE WITH ZANESVILLE LOAM BASE INTENDED FOR HEAVY WORK

FIG. 14 MACHINE-MADE CORE FOR BRASS-FOUNDRY WORK

gives the core more vent than it appears to have. The relatively fine Del Ray bank sand serves to close up the spaces and prevent the brass from entering the core. The combined flour and oil bond burns out at a relatively low temperature, allowing the core to crush before the metal and also facilitating the cleaning of the core.

77 For cast-iron radiators the cores must have a great strength and at the same time vent with remarkable freedom. Many radiator companies are using relatively coarse sands in their mixtures. A microphotograph of a section of radiator core made from a coarse New England sand is shown in Fig. 15. This sand, however, carries a good deal of fine material with it which with the oil forms a network between the larger grains and to some extent stops the passages. The percentage of small material, however, is not sufficient to reduce the vent near the danger point. The firm using this grade of sand has installed a mill for washing the old sand and permitting its re-use in the cores.

78 Fig. 16 shows a microphotograph of a section of a core made entirely from the washed burned sand. The finer material is all washed away but each grain of sand is covered with a layer of oil so that when a core mixture is made the grains are primed, as it were. The fact that the grains already have an oil covering and that there is no fine material in the washed sand to absorb oil, has made it possible to produce stronger radiator cores from washed sand with the same percentage of oil that is used in the new and unwashed sand cores.

79 The core shown in Fig. 17 is made from gangway sand with glutrin as a binder. The glutrin sweated to the surface of the core, forming a hard skin, indicated by the darker area near the outer edge of the core. The interior of this core was so soft that it would readily crush before the metal. These cores were used to form finished faces for bearings in agricultural machinery, the cores being knocked out of the castings, the hole cleaned and the parts assembled without machining. The fact that the glutrin sweated to the surface made a hard close surface which acted almost as a chill and produced a remarkably smooth casting. The rotten interior gave free vent. At another plant in the immediate vicinity some cores were made from the same sand with linseed oil as a binder but the binding ratio had to be very low, that is, a large amount of oil used to bind a given amount of sand. This was also true in the case of the glutrin.

80 *Reactions between the Sand and the Binder.* A quantity of gangway sand was shipped to the laboratory at Covington and together with other sands that had been giving trouble in different parts

FIG. 15 RADIATOR CORE MADE FROM NEW BANK SAND

Showing how fine material holds the oil on the surface of the pebbles away from the contact points of the grains.

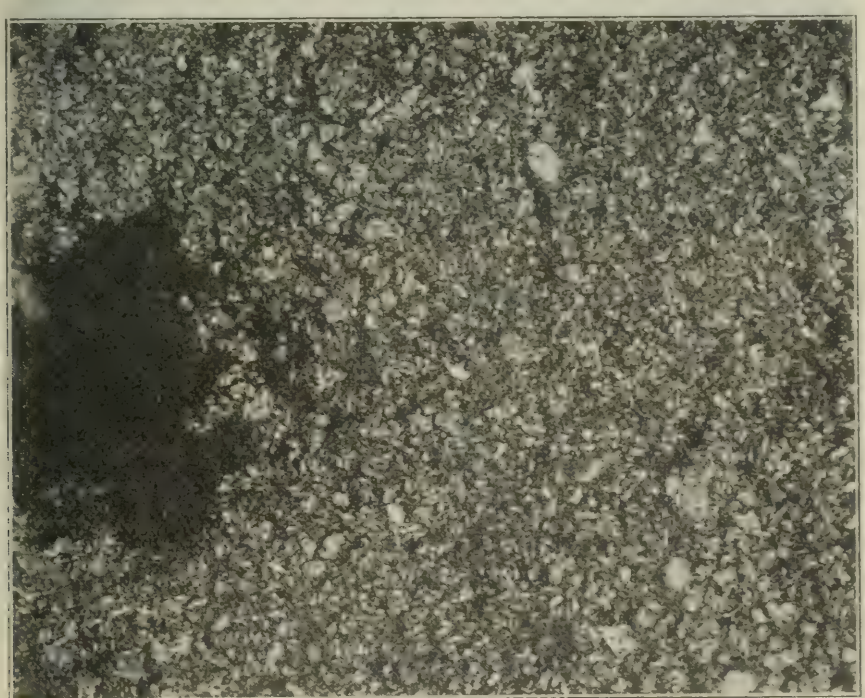


FIG. 16 RADIATOR CORE MADE FROM BURNED SAND

This sand was black and covered with charred oil (differing from that in Fig. 15) which makes the core appear to have less vent than that shown in Fig. 15.

of the country a series of experiments was started. From the behavior of these sands it was evident that some action was going on which destroyed the binder, other than the mere absorption of it, which takes place when the oil unites with clay and is rendered inoperative.

81 To test the correctness of this theory a batch of gangway sand and a batch of Michigan City sand were measured and placed in two bottles. As glutrin was wholly water soluble it was chosen as a binder. Three equal batches of glutrin were diluted with a given amount of water. One was placed on the Michigan City sand,

FIG. 17 CORE MADE FROM GANGWAY SAND WITH GLUTRIN AS A BINDER, SHOWING THE SWEATING OF THE BINDER TO THE SURFACE

another on the gangway sand, and the third held in reserve. After leaving the binder several hours each sand was leached with water until all the binder was washed out. The two batches of glutrin were then concentrated by boiling to expel the excess water. They were brought to the same volume as the batch held in reserve, and three sets of cores made, one from the glutrin which had been on the gangway sand, one from that which had been on the Michigan City sand, and the third from that which had not been on any sand.

82 There was very little difference in the strength of the cores from the second and third batch, but the cores made from the glutrin

which had been on the gangway sand were not one-fourth as strong as the others, showing that the sand had destroyed the bonding power of the glutrin. A few tests soon showed that it was alkaline and the addition of a small amount of acid destroyed the alkalies.

83 Following up this clue, two batches of cores were made from gangway sand, in one case the sand having been treated with acid to neutralize the alkalies and in the other it was left in its natural condition. The acidulated sand produced cores more than four times as strong as the other batch. Continuing the investigation, experiments were made by treating with acids a number of sands giving poor results with different binders and it was found that neutralizing the alkalies with acids increased the strength of the cores greatly. This is true both in the case of glutrin and oil.

84 Since these experiments the subject has been taken up with a number of foundrymen and several exceedingly interesting cases brought to light. In one instance a foundry was purchasing city water of a very pure grade at a relatively high cost. To avoid this expense an artesian well was sunk. Immediately trouble appeared. The cores cut and washed and it took some time to trace the difficulty. Finally the city water was again used and the trouble disappeared. The water from this well was clear, sparkling and tasted good, but it was high in lithia salts and evidently contained ingredients which combined with the binders to destroy them.

85 Alkalies tend to saponify oil, thus destroying its bonding power. They also seem to act on resin, glutrin, and some other binders. In the case of the large manufacturing concerns using a great many oil-sand cores it would probably pay to purify the water for the core room just as the feed water is purified for the boilers.

86 *Determining the Active Bond in Clay.* Formerly when a chemist received a sample of sand for analysis he determined the alumina, figured it as kaolin, and called it "bond." From various results with bonded sands it became evident that this free and easy method was not sufficient, and some way of differentiating between a fat and a lean clay had to be found. Several firms and different departments of the Government were making use of a test for colloidal matter. Chemists have long recognized a series of amorphous bodies known as colloids, or as one noted chemist writing in popular vein recently says, what in the laboratory are called "messes."

87 When matter is in one of the colloidal forms it has intimately associated with it a large amount of water, forming what is termed a "gel." The soluble colloids are called "sols." Certain clays are

far more plastic and have greater binding powers than others, and it has always been known that heat destroyed this binding power. It is now certain that in these clays at least a portion of the alumina is in the colloidal form. For decades the best clays have been found in the secondary deposits where they had been associated with organic matter, but these properties have been found even in the old fire-clays of the coal measures. When first exposed, these are in a compact form much of the water having been expressed or driven out by pressure, and when exposed to the weather they absorb moisture.

88 A splendid example of colloidal clay is found in the Mississippi River. The Ohio and the upper Mississippi both flow through old rock formations while the Missouri and particularly the north and south branches of the Platte flow through more recent geological formations. These formations are rich in a highly colloidal clay, which when it comes into suspension in water remains indefinitely. Engineers had long puzzled over the fact that the water at the mouth of the Mississippi became clear as soon as it left the mouth of the river, but the explanation is exceedingly simple. The salt water of the sea destroys the colloids and allows the mineral matter to settle rapidly. Those familiar with the old Mississippi steamboat days will recall that the water supply for those boats was obtained by pumping the muddy water into tanks and introducing a little alum. The colloids were thus destroyed and the mud quickly settled.

89 The only test for colloids of which there is any record is the one used by several departments of the Government and a number of firms in this country, and is founded on experiments carried on in Germany.

90 The aniline dye known as malachite green is used for this purpose. A given amount of malachite green is weighed out and dissolved in 400 or 500 cu. cm. of water. Into this from 10 to 20 grams of the sand or clay to be examined is introduced and the whole is shaken in a shaking machine for an hour. The bottle containing the material is then taken out and the solid matter allowed to settle to the bottom. A portion of the liquid above the solid matter is drawn off, placed in a color tube, and compared with the dye diluted with the same amount of water which was used in making the original solution.

91 The amount of dye absorbed by the colloidal matter in the sand indicates the bond present in the sand. Table 1 shows a series of colloidal readings on a group of sands examined under the direction of the author. A sample of kaolin was taken as 100 per cent, and other sands were compared by the amount of dye they would

absorb. A number of clays have since been found that run considerably over 100 per cent.

92 This scale has served to compare the different sands in the laboratory but before a method of this kind becomes general throughout the country some standard should be fixed. The readings given in the table represent only a small portion of the determinations made. It will be noted that the materials may be grouped into three classes. The first three samples tested represent washed silica sands with approximately the same colloidal reading; the next five represent beach or lake sands from the shores of the Great Lakes. All of these are wind driven sands with about the same colloidal reading.

TABLE 1 COLLOIDAL READING ON SANDS

Name	Bond No.
Washed silica core sand, Bethlehem, Pa	2.17
Washed Ottawa silica sand, Ottawa, Ill	2.30
Silica sand, Derby, England	2.50
Sharp sand, Faribault, Minn	5.02
Core sand, Sauk Center, Minn	5.21
Michigan City core sand	5.27
Crystal Beach lake sand, Buffalo, N. Y	5.30
Beach sand, Buffalo Am. Rad. Co	5.43
Core sand, Falkirk, England	10.66
Core sand, Falkirk, England	10.81
Manistee lake sand, Flint, Mich	10.70
Old molding sand, Faribault, Minn	10.80
Silica core sand, Duquesne Steel Fdy. Co	10.84
Mansfield core sand, Birmingham, England	15.38
Bank sand, Rochester, Mich	21.63
New molding sand, Faribault, Minn	32.60
Gangway sand, Moline, Ill	43.50
Lumberton sand, Hainesport, N. J	53.43
Kaolin, Fimer and Amend	107.60

93 These are followed by five core sands which would ordinarily be considered as fairly sharp. The first two are English core sands; the third is the Manistee sand as used at Flint and Detroit, Mich. The reading on the old molding sand given fourth is not to be depended upon for its original colloidal reading, since the use of a sand always changes the colloidal content, the colloids themselves being usually destroyed. Burnt sand, however, generally contains material which will destroy the dye and so give an apparent colloidal reading. The old molding sand really had very little bond, though the colloidal or dye reading would place it in a class having a fair

amount. Another case of a burnt sand is given near the bottom of the table, this being the same gangway sand from Moline, Ill., with which the tests in acidulated sand were first carried on. There was absolutely no colloidal matter in this sand. This and other tests led to the belief that the dye reading was to be depended upon when applied to new sands in the condition in which they are dug, but does not apply to material which has been exposed to molten metal. The very high colloidal reading of Lumberton sand is to be expected, as this contains a very large percentage of clay.

94 The behavior of many of the sands tested when made into oil-sand cores showed very conclusively that the dye test gives a good idea as to the amount of oil which would be destroyed by the sand. Apparently the dye reading gives not only the colloidal matter which will destroy the oil, but in the case of old or burnt sands gives a fair indication of the amount of alkali or other material present which would destroy the oil. This was noted in connection with the gangway sand from Moline. Further tests will be made along the line of examining sands and studying the relationship between the natural bonding power and the colloidal reading as given by the dye tests. Thus far this seems to be the best test found.

95 *Action of other Binders on Oil.* When an oil is used as a bond it is not only destroyed by any clay or colloidal matter present but may also be destroyed by other ingredients used as binders. Where green strength is required in a core, foundrymen frequently introduce flour or dextrine into oil-sand mixtures. The oil first combines with the flour to form an oil-flour paste which has very little bonding power, but when so held it is not in a position to act as a binder between the sand grains. In consequence what happens in a mixture of this kind is that a portion of the oil unites with the flour and is itself destroyed for binding purposes. At the same time it renders the flour less efficient than it would be had it been mixed with water. The balance of the oil can act in its usual manner between the sand grains.

96 *Testing Binding Power of Liquid Binders.* The final binding power of any compound is measured by the solid bond left in the baked core. Paint chemists test their oils by drying them down to a film and seeing what percentage of weight has been lost in this action. In the case of Table 2 the binders shown in the column at the left were each weighed out into porcelain crucibles and first placed in an air bath and subjected to a temperature of 100 deg. cent. or 212 deg. fahr. for 24 hours and then weighed. These results are entered in the second column. The crucibles were next placed in the core oven at

a temperature of 400 deg. for 1 hour and again weighed and the results indicated in column three. Some of the crucibles were then returned to the core oven for another hour, exposed to a heat of 410 deg., cooled, and again weighed and the results recorded in the fourth column. The samples were then all heated in the air over a blast lamp to burn off the carbon, and the percentage of ash is recorded in the fifth column.

TABLE 2 DETERMINATION OF SOLIDS IN LIQUID CORE BINDERS

Name	Dried 24 Hours at 212 Deg. Fahr.	Dried 1 Hour at 400 Deg. Fahr.	Dried 1 Hour at 410 Deg. Fahr.	Burned to Ash	Remarks
Glutrin	51.47	41.98	.....	8.03	Skin
Raw linseed oil	100.22	96.98	95.22	0.22	No skin
Boiled linseed oil	93.49	90.23	.....	0.54	Skin
Soya bean oil	100.91	96.45	94.26	0.65	No skin
Fish oil	100.52	92.70	91.25	1.20	No skin
Paraffin oil	72.05	33.36	10.40	0.11	No skin
Corn oil	100.72	95.35	93.12	0.42	No skin; crawled
Cottonseed oil	100.68	96.08	94.10	0.07	No skin
Light tar oil	31.18	11.65	.....	0.09	Slick crawled
Heavy tar oil	67.21	38.27	.....	0.06	No skin; crawled
Resin oil	62.89	19.59	.....	0.06	No skin; crawled
Crude tar oil	51.83	34.32	.....	0.06	No skin; crawled very much
Resin oil	63.64	10.12	.....	0.03	No skin; crawled
China wood oil	102.20	98.75	.....	0.03	Skin

97 Under the head of Remarks a statement is given as to the behavior of the liquid. Most of it dried down to a skin. The first exception to this was raw linseed oil, and the reason was that even when painted as a film on wood it takes at least 4 days for it to dry. There was evidently not a sufficiently free admission of air in the oven to dry down the bulk of oil in the crucible.

98 *Drying of Raw Linseed Oil.* The appearance of the sample of raw linseed oil after it had been in the core oven 1 hour at 400 deg. is shown in Fig. 18. Had this oil been evenly distributed through a body of sand there would have been sufficient circulation through the body of sand completely to oxidize it. These tests, however, show that while raw, linseed oil is one of the strongest binding oils we have; it is nevertheless a very slow drying oil, and hence pure linseed oil cores would probably take longer to dry than those made from other oils.

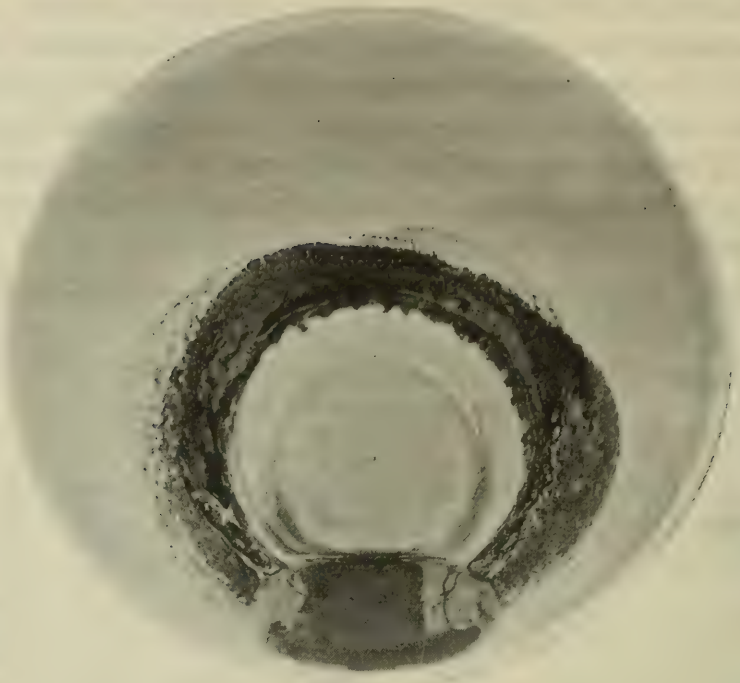


FIG. 18 RAW LINSEED OIL TESTED FOR DRYING QUALITIES

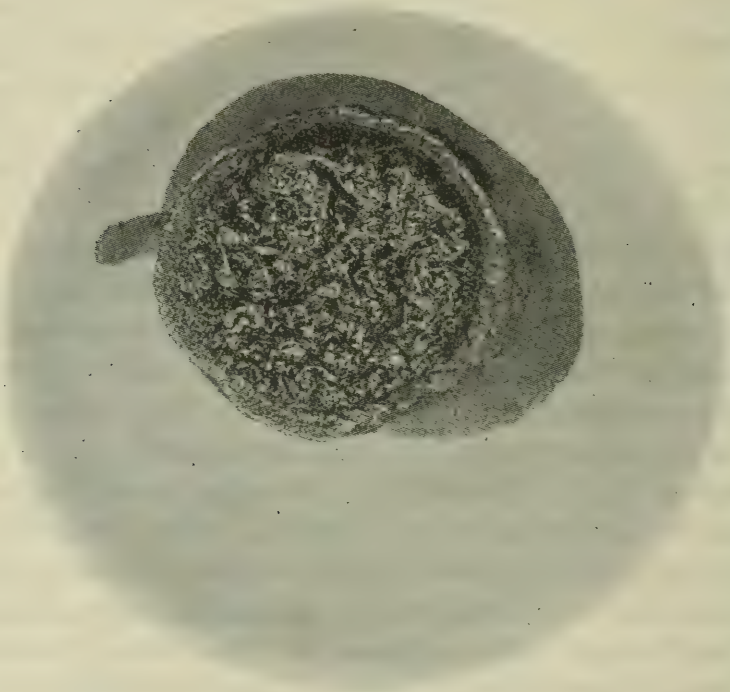


FIG. 19 CHINA WOOD OIL TESTED FOR DRYING QUALITIES

99. *Drying China Wood Oil.* The appearance of the surface of the skin in the case of some of the oils after test No. 2 is interesting. Fig. 19 shows the China wood oil made from the tung nut which grows in China. This is a relatively new oil in this country, and partakes at once of the nature of an oil and a varnish gum, being to some extent a natural varnish. It contains its own driers, so that it dries down fairly readily. It will noticed under the tests in the second column that China wood oil gained 2.2 per cent in weight by oxidation when exposed to a temperature of 212 deg. fahr. Several of the oils gained slightly, but China wood oil leads them all in this respect. It also showed the greatest percentage of weight after one hour in the core oven and the film was strong and tough. When tested as a binding oil it gave excellent results.

100 *Drying of Light Tar Oil.* The light tar and a number of similar oils crawled up the face of the crucible and dried down to a hard glazed film. The crucible containing the light tar oil as photographed after the test recorded in column 2 is shown in Fig. 20. This oil had left only 11.65 per cent of its original weight as a bonding medium. Such oils are used as blending oils in liquid core compounds.

101 *Drying of Boiled Linseed Oil.* The boiled linseed oil dried down to a film in the air bath as would be expected, on account of the fact that it contains artificial driers. The appearance of the film of the boiled oil after the test recorded in column 3 is shown in Fig. 21. Formerly the term, boiled oil, meant linseed oil which had been boiled in open kettles exposed to the air so as partially to oxidize it and certain mineral oxides were added to it. Such oil when used for mixing paints dried down to a film much more rapidly than raw oil. Today practically all of the boiled oil on the market is a compound which has been heated to a certain temperature below that of the boiling point of linseed oil and had various so-called driers added to it. which chemists call catalysts. That is, they are in this case bodies having the property of taking oxygen from the air and delivering it to the oil to hasten its oxidation or drying. The boiling of linseed oil in a kettle or the making of so-called boiled oil by adding driers hastens its setting or drying by partially oxidizing the material. Hence a lower ultimate bonding power would be expected from such an oil, and results seem to show that this is true. For this reason, when considered purely from the standpoint of an efficient bond a raw oil is better than a boiled one.

102 *Effect of a Sticky Oil on Piecework.* The author ran across an interesting condition where the pieceworkers in a core room claimed

FIG. 20 LIGHT TAR OIL TESTED FOR DRYING QUALITIES

FIG. 21 BOILED LINSEED OIL TESTED FOR DRYING QUALITIES

that they could not make a good output when using raw oil, since it gummed the boxes more rapidly than boiled oil. The chemist in charge stated the reason was that the raw oil contained mucilaginous "foots" which had been taken out of the boiled oil. Several oil chemists claim that in the modern process of boiling oil everything in the raw oil is to be found in the boiled oil, but that the mucilaginous material may be partially eliminated by combining it with driers.

103 *Linseed Oil "Foots."* Practically all linseed oil is now made by the hot pressing process, and as it comes from the press contains some exceedingly fine meal, which settles to the bottom of the oil storage tanks and is known as foots. After the clear oil has been drawn off that at the bottom containing the foots is used as a core oil by many foundrymen. It has high binding power but tends to stick to the boxes.

104 *Comparison of Molasses and Glutrin Films.* Before leaving the subject of the tests made to determine the amount of binding power in the various materials, it may be well to illustrate the films left by glutrin and molasses. Fig. 22 shows the glutrin film which dried down to a compact mass very much like the oils. This explains why glutrin cores do not swell in baking, but retain their size and shape so that the baked core if it has been properly supported during baking can be returned to the box after drying and will fit perfectly Fig. 23 shows the film left by molasses. The liquid boiled to the top of the crucible forming a rough mass composed of bubbles. This is why molasses cores are frequently distorted during baking.

105 *Fineness Tests on Sands.* It has already been stated that the size of the grains of sand should be as uniform as possible if the core is to have the maximum amount of venting space. The various sands examined were all tested for fineness and the results given in Table 3. The amount recorded in each column is that which remained on the sieve of the size given at the top of the column. The percentage of material passing through the 100-mesh sieve is recorded in the column headed, 100†.

106 The fineness number given in the righthand column is arbitrary, and is intended for the comparison of different sands. It was reached in the following manner: The percentage which remained on each sieve is multiplied by the number of the sieve through which it passed, with the exception of that which remained on No. 20. This is also multiplied by 20 so as not to introduce a great error. These products are all added and divided by the amount of sand taken which gives the fineness number.

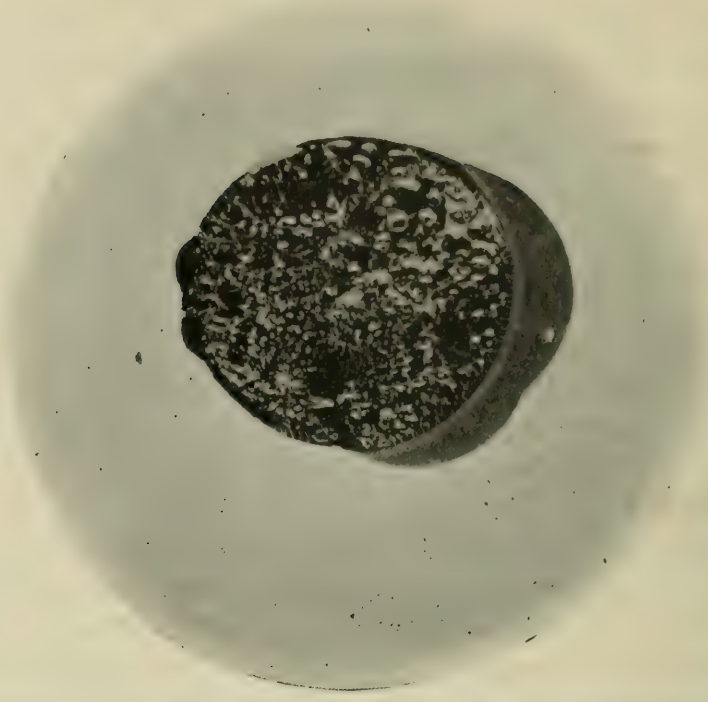


FIG. 22 GLUTRIN DRIED DOWN TO A FILM, SHOWING NO SWELLING

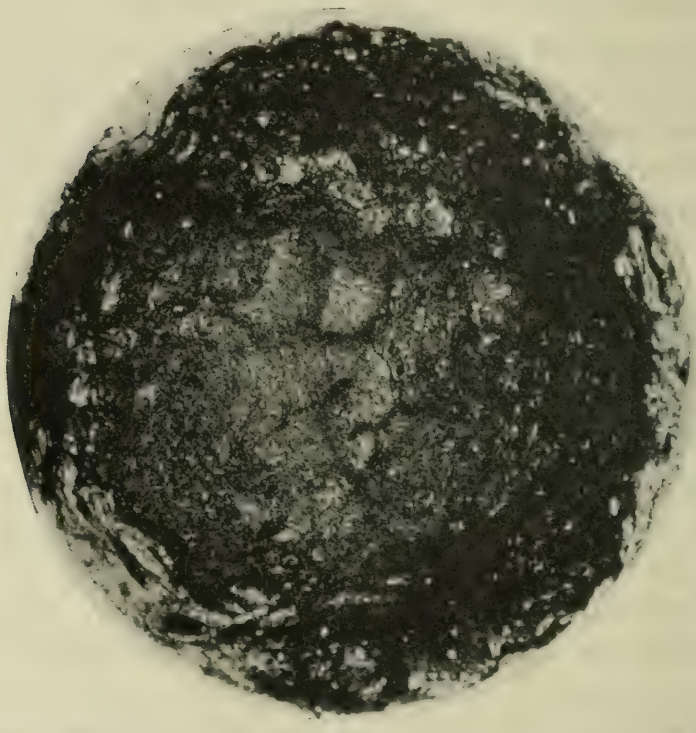


FIG. 23 MOLASSES DRIED DOWN TO A FILM, SHOWING HOW IT SWELLED TO FILL THE ENTIRE CRUCIBLE

TABLE 3 FINENESS TEST

Sand	Mesh						Fineness Number
	20	40	60	80	100	100†	
New England bank	3.7	11.6	17	11.6	13.7	40.6	68
Rockaway Beach	0.7	14.5	46	23.5	11	2.7	47
Michigan City core	0.5	9.8	41.5	35.75	9.55	1.8	50
Coarse bank, Cleveland	6.3	59.6	22.3	5.0	2.6	3.1	30
Clay, Cleveland	2.8	30	11.0	4.9	10.3	41.7	64
Washed silica core	0.1	1.7	30.7	38.0	22.3	6.7	60
Lake Manistee, Mich	0.2	4.3	9.8	12.3	12.4	60.7	83
Bank, Rochester, Mich	0.1	45.7	50.2	3.3	0.1	0.9	32
Strawbridge, England	2.0	16.5	9.6	10.2	22.7	40.0	72
Mansfield, England	0.3	10.1	10.3	10.6	23.7	42.5	74
Burnt, Mansfield, England	0.1	6.3	7.2	10.7	27.9	43.6	77
Gangway, Moline, Ill	1.8	7.9	10.9	9.6	12.5	55.7	78
Core, Pittsburgh, Pa	5.3	42.6	37.0	5.1	4.4	5.0	27
Bank, Buffalo, N.Y	3.1	32.0	38.5	11.8	5.2	8.2	42
Lake, Buffalo, N. Y	0.1	12.6	33.7	24.4	21.6	7.3	55
Core, Buffalo, N. Y	6.3	71.8	20.1	0.8	1.0	0.1	25
Falkirk, England	7.6	32.3	32.5	12.9	7.7	5.0	40
Core, Faribault, Minn	0.2	2.5	38.2	33.4	17.6	7.1	57
Silica, Ottawa, Ill	0.1	72.8	21.6	3.1	0.8	0.7	26
Derby, England	2.5	8.3	46.5	27.3	10.5	3.1	49
Lumberton, Philadelphia, Pa	9.0	34.5	24.1	9.1	12.6	9.1	41
Core, Zanesville, O	6.0	44.0	2.7	9.1	15.4	20.0	49
Molding, Zanesville, O	41.1	31.5	15.2	3.3	1.7	6.5	30
Molding, Zanesville, O	0.6	1.2	0.4	0.3	1.1	95.1	97
Washed silica, Millington, Ill	0.1	56.7	23.8	6.5	5.7	6.7	36
Magnesia, Millington, Ill	4.3	48.6	20.7	5.8	6.8	13.4	41
Core, Sauk Center, Minn	2.4	6.0	19.2	19.5	17.5	33.9	69
Lumen Bearing Co.'s lake	0.1	4.35	19.1	34.05	31.2	10.75	58
Del Ray bank, Detroit, Mich	1.5	0.55	5.1	5.75	21.15	55.05	76
Dolly Ann, Covington, Va	9.4	32.6	26.9	16.2	6.9	6.8	41
Providence River	0.5	2.2	6.5	6.5	13.5	69.8	88
Rockaway Beach	0.1	8.7	34.6	36.7	14.2	5.1	54
Indiana bank	0.1	6.6	44.8	33.6	12.6	1.6	50
Magnesia core, Millington, Ill	0.1	46.1	25.2	7.2	1.2	19.3	44
Yellow, sharp	0.1	30.5	48.5	13.9	2.1	3.6	39
Bank, Cleveland, O	12.7	26.4	27.3	7.2	6.0	20.2	48
Light molding, Cleveland, O	13.8	25.5	9.2	4.5	3.8	42.0	59
Travers City, Mich	2.0	38.4	42.3	9.2	3.5	3.2	36
Silica dust	0.7	1.2	14.1	10.3	9.9	62.3	82
Sugar, Cleveland, O	17.1	36.2	23.2	11.5	5.3	5.5	37
Del Ray bank, Del Ray, Mich	0.2	0.9	5.3	10.5	25.1	57.1	86
Gravel, Cleveland, O	31.5	24.3	14.2	6.2	7.2	15.4	42
Bank, Cleveland, O	6.0	10.8	14.8	11.1	21.1	34.4	67
Gravel, Cleveland, O	41.8	32.1	20.5	3.3	0.2	1.0	26
Sand blast, Cleveland, O	2.2	7.9	12.6	9.8	13.5	52.8	76
Silica, Cleveland, O	1.4	5.5	17.7	7.8	19.8	57.0	86

TABLE 3 FINENESS TEST—Con.

Sand	Mesh						Fineness Number
Silica, Cleveland, O.	3.8	37.8	40.4	10.3	4.2	2.4	36
Sharp, Cleveland, O.	2.6	5.6	29.3	39.1	19.8	3.5	56
Bank, Chicago, Ill.	0.1	7.4	40.4	28.5	19.2	4.0	54
Custer Park, Custer Park, Ill.	0.3	5.9	29.8	19.3	18.2	24.7	64
Silica, Cleveland, O.	0.1	6.1	42.4	36.2	13.3	1.7	52
Bank, Cleveland, O.	4.1	13.2	18.1	13.3	19.0	31.7	52
Silica, Cleveland, O.	0.1	54.4	34.0	5.9	3.1	2.0	33

107 It is interesting to note that most of the sands adopted by foundrymen through experimental work have very nearly uniform sized grains, or at least will leave more than half of the sand on two consecutive sieves. For instance, the Michigan City core sand, the third in the column, has over 75 per cent of its sand on the 60 and 80-mesh sieves; the washed Ottawa silica sand has 72.8 per cent of its grains on the 40-mesh sieve, and 21.6 per cent on the 60-mesh sieve, or over 94 per cent of the sand on two consecutive sieves. The Del Ray bank sand so extensively used by brass foundries, is exceedingly fine, 55 per cent of it having passed the 100-mesh sieve. As seen under the microscope the grains are clean and rounded so that it has considerable vent. One of the fine sands used in the East is the Providence River sand, and of this 69.8 per cent passed the 100-mesh sieve.

108 *Specifications for Core Sand.* More investigation along this line and the coöperation of manufacturers will result in specifications for core sands for different classes of work. These specifications should include the relative fineness, the percentage of bonding material which will destroy oil, as determined by the dye test for colloids or some other test to be determined later, and the percentage of alkalies or lime.

109 *Mineral Composition of Sand Grains and its Effect.* A good core sand should be free from shale or limestone pebbles if it is to be used with oil, since the shale is liable to form a fluxing ingredient. For some classes of grey iron work, sands carrying a considerable percentage of limestone pebbles are used. These sands cannot be used with oil, but can be used with pitch compounds or glutrin. The first time they are exposed to molten metal, however, the lime which is next to the casting will be burned to quick lime. This will result in a considerable volume of carbon dioxide gas which must pass out through the vent, but the burning of the limestone to quick lime will partially disintegrate the core and make it easy to clean. If any of this old core sand is used in the new mix, the quick lime contained in

it becomes a rather efficient binder, acting much as does flour or starch in that only the portion of the lime in contact with the sand grains is affected, the balance of it simply stopping up the vent.

110 At least two large foundries are making extensive use of a core-sand mixture containing heavily bonded loam sand and a quantity of limestone pebbles. For some classes of very heavy work no artificial binder is used, the clay in the new sand and the quicklime in the old sand being depended upon for that purpose. It is necessary in such cases, however, to give the cores a coat of blacking in order to peel the castings, and in making up the cores a large amount of coke has to be used to form vent passages.

111 *Effect of Moisture on Volume of Sand.* In experimenting with a wide range of binders several sands representing different types were selected as standards and cores made with each series. The Ottawa silica sand, Michigan City sand, and the Rockaway Beach sand, were three types of sharp sands, or sands carrying comparatively little bond, selected. One day the man making the cores reported that from a measured amount of sand he was obtaining more cores than formerly. Upon investigating it was found that he had taken one batch of sand from the bottom of a bin of Michigan City sand where it was wet and the next from the top of the adjoining bin where it was dry. To see if this would explain the matter a quantity of dry Michigan City sand was measured, 10 per cent of water by volume added to it, the mass mixed over by hand, and an attempt made to put it back in the same measure from which it had been poured. It was found that 40 per cent could not be returned to the measure. Experiments were then made by taking measured amounts of sand and adding first 1 per cent and then 2 per cent of water, and so on up to 20 per cent or more. It was found that at first each addition of water increased the volume of sand until about 10 per cent had been added, and that beyond this the volume began to decrease once more, until the volume of water was so great that the sand could settle out of it, when the volume of sand became very nearly the same as that for dry sand.

112 The explanation is simple. In dry sand the grains slide past each other until the maximum number of bearing points are in contact. When there is just sufficient water present to form a film over each grain of sand, two grains coming in contact adhere and will not move further without the application of force. Even fairly hard ramming will not drive the sand back to its former volume. This condition is graphically illustrated in Fig. 24. On the left there are two

graduated cylinders, one containing 500 cu. cm. of dry Michigan City sand and the other 50 cu. cm. of water. On the right is shown the large cylinder with the wet sand in it, and the 10 per cent of water having been mixed with the sand by hand and the material dumped back into the cylinder. It filled the cylinder up to the 780 cu. cm. mark, an increase of 270 cu. cm. or over 50 per cent. In putting the sand in, the cylinder was rapped down on the hand as hard as pos-

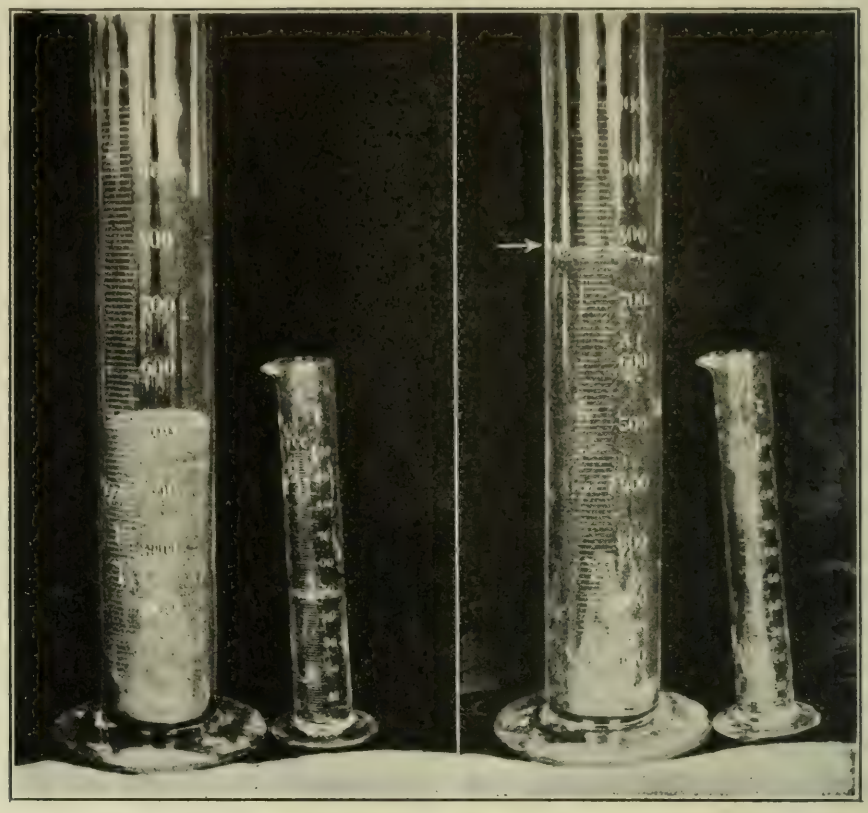


FIG. 24 CHANGE IN VOLUME OF SAND CAUSED BY THE ADDITION OF 10 PER CENT OF WATER

sible without danger of breaking it. The results of experiments with New England bank sand and Rockaway Beach sand are given in Tables 4 and 5.

113 The practical application of this is that if the men in the core room are allowed to take sand as it comes and measure it by volume they will be getting constantly varying ratios between the sand and the binder. If a given volume of dry sand is taken it will contain a certain number of grains and there will be a certain ratio to the binder. If the same volume of wet sand is taken there will

be a very much smaller number of grains and hence a larger proportion of binder or smaller binding ratio, which will result in a stronger core, provided the mixing is properly done.

114 All core sand should be dried before the binding material is mixed with it. If the core sand is measured by weight the presence of moisture in it also introduces an error. Fortunately these errors tend to make stronger cores than would result from the use of dry sand, on account of the decrease in the bonding ratio, but this increase in the proportion of binder is an unnecessary and increasing expense which must be borne.

TABLE 4 TESTS WITH NEW ENGLAND BANK SAND FROM WATERBURY, CONN.

Sand, by Volume	Water, by Volume	Water by Weight, Per Cent	Sand Water Added, Volume	Increased in Volume, Per Cent
500	5	0 77	590	18
500	10	1.54	680	36
500	15	2.31	750	50
500	30	4.62	790	58
500	45	6.93	830	66
500	60	9.24	810	62
500	75	11.55	810	62
500	90	13.86	800	60

TABLE 5 TESTS WITH ROCKAWAY BEACH SAND, ROCKAWAY BEACH

Sand, by Volume	Water, by Volume	Water by Weight, Per Cent	Sand Water Added Volume	Increase in Volume, Per Cent
500	5	0.68	700	40
500	10	1.36	740	48
500	15	2.04	730	46
500	30	4.08	700	40
500	45	6.12	810	62
500	60	8.16	810	62
500	1000	Flooded	470	..

115 *Effect of Damp Sand on Hand-Mixed Oil-Sand Cores.* In connection with the use of damp sands an interesting core is shown in Fig. 25. This was made from sharp Lake sand and oil at the ratio of 50 to 1. The sand was fairly damp when taken from the bin, containing at least 3 per cent of moisture by weight. The sand was spread on the coremakers' bench, the oil measured and poured over it, and the batch thoroughly mixed by hand, until the oil appeared to be evenly distributed. A little water was then added to temper it

thoroughly and again rubbed over two or three times by hand. It was not passed through a riddle, however.

116 A batch of cores were made from it and baked. When they came from the oven small white spots of unbonded sand appeared on the surface and the sand could be poured from these. The hole shown in Fig. 25 is less than $\frac{1}{16}$ in. wide and a little over $\frac{1}{8}$ in. high. It extends into the core for $\frac{3}{8}$ in. A number of these were seen all over the surface of the core.

117 Another batch of sand was mixed in the same way, and passed through a No. 4 riddle once. Cores were made from this and baked. They were found to contain little spots where a dozen or more grains of sand had adhered and the water on them had prevented the oil from covering them. These spots of dry sand fell out leaving small spots on the face of the core and upon breaking the same condition was found to exist through its entire body. A portion of the same batch was put in a mixing mill and ground for 2 minutes. Cores made from this showed perfectly uniform results.

118 These experiments clearly showed two facts: first, damp sand should never be used in making up core mixtures, and second, core sand should be ground or thoroughly mixed to insure the incorporation of the oil with the sand and the covering of every grain of sand with the oil.

119 *Binders Tested.* A partial list of the binders tested is given in Table 6. This contains nothing but materials from which binding compounds are made or materials used as binders, without being incorporated with other materials to form core mixtures. The name of the one donating the sample is shown at the right. In addition, a number of other makers of core compounds have furnished samples of their standard products. In this investigation, however, the underlying principles which govern the bonding of core sands are sought and hence, so far as possible, known compounds are used.

120 The results of one series of tests in comparing some of the binders is shown in Fig. 26. The paraffine oil and crude oil which are frequently used in core oils for blending show no binding power whatever, or at least they broke at less than 1 lb. per sq. in. The black compounds tested in this series were not suited for use with the Rockaway Beach sand, but should have been used with those containing some loam. The pitch referred to is what is known as parolite, a product furnished by the Standard Oil Company. A sample of raw linseed oil was also tested in this series, but could not

be broken on the testing machine then in use. Since this time experiments have been tried and oil sand cores made at the ratio of 1 to 50, breaking as high as 160 lb. per sq. in. in tension.

SELECTION AND COMPOUNDING OF CORE MATERIALS

121 *Cores for Steel Foundry Work.* The selection of the proper materials for use in any given core room depends upon local conditions as to supply of core-forming materials and the product being manufactured. A wider variety of materials can be used in connection with grey iron work than in most other classes of work.

FIG. 25 MICROPHOTOGRAPH OF CORE SHOWING HOLE WHERE UNBONDED SAND FLOWED OUT

122 Heavy open-hearth steel work requires the best materials. Both the mold and the cores must be made of silica sand and nothing that has a tendency to fuse at low temperatures can be employed. At the same time the cores must be of such a nature that they will crush readily before the metal as it shrinks. This has brought into use many different mixtures all having the same object in view. Flour or flour and molasses are sometimes used as binders and the cores baked until the bond is practically destroyed so that the core is rotten. Core arbors and core rods are relied on to hold the mate-

rial together and the surface is thickly coated with a silica wash which gives a strong skin to the core. Such a core will usually resist the action of the molten metal and crush readily when the casting shrinks.

TABLE 6 SOME OF THE BINDERS TESTED

NAME	SHIPPED BY
Boiled linseed oil	Spencer Kellogg and Sons Buffalo, N. Y
Raw linseed oil	Spencer Kellogg and Sons, Buffalo, N. Y
Fish oil	Young & Kimball, Boston, Mass.
Paraffine oil	Standard Oil Co., Cleveland, Ohio
Corn oil	Standard Oil Co., Cleveland, Ohio
Cottonseed oil	Standard Oil Co., Cleveland, Ohio
Parolite (pitch)	Standard Oil Co., Cleveland, Ohio
Light tar oil	Atlantic Turpentine Co., Savannah, Ga.
Heavy tar oil	Atlantic Turpentine Co., Savannah, Ga.
Resin oil	Atlantic Turpentine Co., Savannah, Ga.
Resin oil	Pensacola Tar & Turp. Co., Gull Point, Fla.
Molasses	Hill & Griffith Co., Cincinnati, Ohio
Flour	Hill & Griffith Co., Cincinnati, O.
China wood oil	L. G. Gillespie & Son, New York
Blending oil	Sun Oil Co., Toledo, Ohio
Crude oil of tar	Pine Turpentine Co., New York
Silicate of soda	Philadelphia Quartz Co., Chester, Pa.
Foundry dextrine	V. C. Bloede, Baltimore, Md.
Special dark dextrine	Corn Products Refining Co., Waukegon, Ill.
Special canary dextrine	Corn Products Refining Co., Waukegon, Ill.
Filtered whale oil No. 1	Harvey & Outerbridge, New York
Filtered whale oil No. 2	Harvey & Outerbridge, New York
Filtered whale oil No. 3	Harvey & Outerbridge, New York
Special varnish gum	L. G. Gillespie & Son, New York
Michigan resin	Cadillac Turpentine, Co., Cadillac, Mich.
Oil No. 1 filtered and blown whale	Harvey and Outerbridge, New York
Raw pitch	Westinghouse Electric Co., Cleveland, Ohio
Distillery slop	Cutter Wood Supply Co.
Hydrol	Corn Products Refining Co., Argo, Ill.
Starch	Corn Products Refining Co., Argo, Ill.
Starch	B. Remmers & Sons, Bourse Bldg., Philadelphia

123 Others use both clay and sawdust in their core mixtures, the sawdust burning out and allowing for shrinkage and the clay acting as a binder. The burning of the sawdust frees the clay and permits the cleaning of the core. In one light-work steel foundry shop a core made of silica sand, Welsh Mountain clay and sawdust is used, the core being baked until the sawdust is partially charred.

124 For work with great shrinkage core mixtures using a binder like glutrin which will sweat to the surface may be used to good advantage, as a core of this kind has a hard skin and a rotten or soft interior. All such cores, however, have to be baked carefully at a certain temperature for if the skin is burned there is nothing left of the core. Glutrin is used in steel facing sand and in many clay and silica sand core mixtures.

125 *Cores for Brass, Bronze and Aluminum.* The other extreme in the core field is found in the brass and bronze foundry. Here the temperatures encountered by the cores are much lower, and greater diversity of sands and binders can be employed. In a case of this

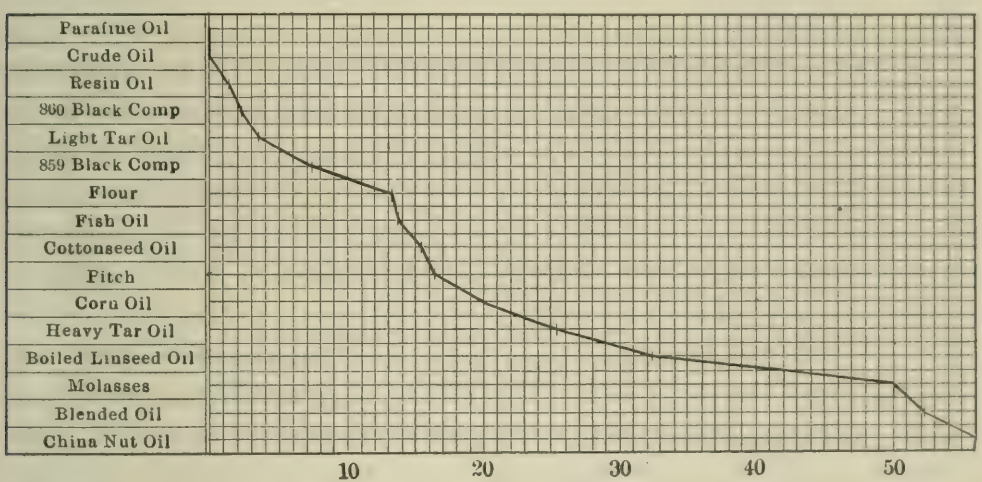


FIG. 26 DIAGRAM SHOWING RELATIVE STRENGTH OF A NUMBER OF BINDERS

kind what is needed is a core with a close surface which is still free venting enough to permit the mold to be filled quickly and the gases to escape readily.

126 Clay is out of place as a binder in brass and bronze work and the material used should be of such a nature as to burn out at a rather low temperature.

127 In dealing with aluminum a metal with maximum shrinkage and minimum hot strength is encountered. This requires a core that yields readily and softens as quickly as the metal strikes it. Two courses are open: to use a mixture giving a core with a soft interior and practically all the binder on the surface so that as quickly as the surface is disintegrated the core can be readily crushed; or to use a binding material which softens as soon as heat strikes it whether it burns or not. Resin is the only binder which fully meets this latter

requirement, and it is frequently used to good advantage in aluminum work. These cores clean out readily if the castings are taken from the mold hot and cleaned out at once, but if they are allowed to get cool the resin cores harden once more and it is almost impossible to get them out.

128 *Cores for Grey Iron and Malleable Iron.* For grey iron or malleable work the core problems to be met are almost infinite. In the radiator shops and in stove foundries producing gas range burners it is necessary to have cores which are exceedingly strong and free-venting and which will stand being entirely surrounded by metal with the exception of a small print. This means a core made from sharp sand and at present all such are of oil or mixtures of oil and glutrin as binders.

129 Oil-sand cores have one advantage possessed by few other binders, that a core made from clean sand and oil has no tendency to absorb moisture and can be kept without losing strength for an indefinite length of time. The writer has cores in his possession, now nearly five years old, made of linseed oil and clean sand, some of which were used after they were four years old and seemed to give as good results as the new ones.

130 A core for the most exacting requirements met by oil-sand mixtures must have a tensile strength of at least 75 lb. per sq. in. In such mixtures it is usually the most economical to employ only high-grade materials and particular care must be taken to see that the sand contains no loam which will destroy the bonding value of the oil. Linseed oil at $1 per gal., as it has been this past year, makes the cost of the old standard binder almost prohibitive and has led foundrymen to look for cheaper materials. All of the paint and varnish oils, including China wood, Soya bean, corn and cottonseed oils have found a place in the core-oil trade. The usual blended oils, on the market also carry resin and neutral oil. As a rule the neutral oil does not add appreciably to the binding power of the compound but it does serve to carry the resin into the mixture and to give a waterproof oil of sufficient strength for ordinary work at a much lower cost per gallon than the old standby linseed. As linseed oil, however, is the ultimate standard of excellence for all core oils, some firms prefer to use it and either to dilute it themselves or to use glutrin in the mixture with it. In many grey iron and malleable foundries flour or dextrine is used as a binder and for some classes of work they give entire satisfaction.

131 *Cost of Strength in a Core.* In solving any core-binder problem the ultimate cost is the guiding factor. To determine the cost of a cubic foot of core several points must be considered: (1) the strength of the core mixture both green and dry; (2) the character of the surface which the core presents to the metal; (3) the question of the percentage of vent area in the core; (4) the ability of the core to resist moisture; (5) the ease with which the core can be cleaned from the casting; (6) the character of the fumes or gases given off during the baking of the core and when the mold is poured; (7) the number of cores which a man can make in a day from a given mixture; (8) the cost of drying; (9) the expense of the rigging involved with the use of a given binder.

132 The strength of the green mixture is important in cases where cores must stand at a considerable height above the plate during drying, and if the core has any overhanging bodies it must possess sufficient green strength to carry these. Green strength may be afforded by using a bonded sand, that is, a sand containing clay, or by using flour, dextrine or some other glutinous or starchy material in the mixture. The strength of the dry core is governed by the amount of pressure it must resist as the metal enters the mold. All cores tend to float as the metal enters and if not sufficiently rigid to resist this tendency, are likely to be broken or displaced.

133 Cores must also have sufficient strength to enable them to resist handling. The strength of core mixtures in use in the foundries in which we have carried on experiments varies from less than 5 to over 100 lb. per sq. in. The proper strength for a given condition can be determined only by experience, but after this has been decided, any other sand or binder mixtures can easily be compared with the standard by suitably testing the new one. Foundrymen have generally adopted a bar 1 in. sq. and broken it on supports 12 in. apart. This is suitable for oil-sand and fairly strong mixtures, but does not give good results for mixtures having low strength. For these the tensile method using a delicate machine is to be preferred or bars 4 or 6 in. long and 1 in. sq. may be made and broken in the middle.

134 Under the second heading must be considered the character of the surface presented to the metal, its ability to carry off the air or gases displaced by the metal as it enters the mold, its ability to resist the cutting action of the metal as it flows into the mold, and the freedom with which the burned core separates from the surface of the interior of the casting. As has already been stated, in the case of metals which have extreme fluidity and a tendency to search out all

small openings in the core, it is necessary to use a mixture which will present a close uniform texture, the voids or spaces of which are so fine that the metal cannot flow into them. In the case of such metals it may be necessary to use some facing or blacking on the core to close partially the spaces or voids and also to make the core clean out more readily. Cores for cast-iron water pipe and a large variety of similar grey iron work are given a coating of blacking and the interior faces of the mold made to peel just as well as the exterior. If the core itself does not present a proper surface to the metal it must be considered whether it would be cheaper to select a material which does or to treat the surface of the core in such a way as to improve it.

135 Under the third heading, the percentage of venting area, must be considered the size of the sand grains and their character; also the character of the bond and the extent to which it blocks the vent. In the case of castings which are entirely surrounded by metal with the exception of a small print, as for instance, gas stove burners, the sand mixtures must have a maximum amount of vent and a minimum amount of gases to be driven out, while in the case of aluminum the metal solidifies so quickly that a very small amount of gases has to be taken care of. The thickness of the casting that is to surround the core also has an influence on this as it determines the duration of the high degree of heat.

136 Under the fourth heading, the resistance to moisture, must be considered the length of time the cores remain in the molds and the manner of handling and storing them. If high-grade oil-sand cores are used, their moisture-resisting properties are such that no ill effects need be feared from leaving them in the mold several hours before it is poured or in connection with their storage, but many other binders show a tendency to absorb moisture. Where this is the case, if there is a considerable saving in the use of one which absorbs moisture, it may be necessary to see if some means cannot be found for so handling the cores as to minimize their exposure to moisture. If they are kept in a dry place and are introduced into the mold only a short time before pouring, there will be no trouble on account of dampness and the saving in the cost of binder may more than compensate for the additional handling.

137 The fifth heading, ease of cleaning, is one of the important problems in the expense of making castings. A proper core mixture should rattle out without any difficulty, leaving a good smooth interior surface. If this is not so, some means must be secured of free-

ing the surface from the metal or of making the entire core of such a nature that it will soften under the action of the heat. In the case of brass and bronze cores, they are very largely blown out by dipping the hot castings into water, the steam formed blowing out the core. Iron castings cannot be so treated, hence the core must be compounded and baked so as to clean freely. One of the most common causes of difficulty in the cleaning room is the use of an excess of binder in the core room, or the use of sands which are too heavily bonded, that is, contain too much clay.

138 Under the sixth heading, the character of fumes, must be considered both the effect of fumes in the core-room and foundry. With regard to the character of fumes, certain oil mixtures give off very disagreeable odors after the cores are withdrawn from the oven and placed in the racks to cool. If less offensive mixtures which are equally efficient can be found it will greatly improve the working conditions of the core-room men; if not, ventilating devices should be provided to draw these fumes away from the coremakers. In the foundry the importance of this question depends to a large degree upon the volume of the cores used for each individual casting. If there are only a few small cores in each mold the volume will not be sufficient to trouble the workman, no matter what binder is used, but if the mold is largely composed of cores it becomes an important item in the comfort of the molders at pouring-off time, a great many castings having been lost on account of careless pouring of a molder in hurrying from the stifling fumes of the surrounding molds. Whale oil and fish oil give particularly bad fumes in these cases, though they form excellent binders. By lighting the gases as they come from the vents in the mold the disagreeable odors can frequently be reduced, but even then they may be sufficient to make a change of binding materials desirable.

139 The number of cores per man per day is influenced by the character of the binder, since if the material tends to stick to the core box, the workman cannot produce as many cores as in a free working material. In many binders the trouble with the sticking was found to be due to too strong a core mixture and as soon as the proportion of binder was reduced the trouble disappeared. In the cost of drying, the length of time it takes to dry each different class of cores when using different binders determines the capacity of the core ovens, their fuel consumption, and the floor space which must be devoted to equipment of this kind. Quick drying core mixtures increase this capacity and so decrease the plant expense.

140 High-grade oil-sand cores made from clear sand and oil have no green binding power and hence they must all be provided with driers which support the core throughout its entire length, or the cores must be bedded in open, clear, sharp sand containing no binder. One of these procedures requires a considerable outlay for driers and the other an expenditure of time for bedding in. If the output from a given core box or pattern is to be limited, it may pay better to use a mixture having some green bond, so that the cores may be allowed to stand on one end on the plate or be supported in some other convenient manner which will avoid the making of driers. If very large outputs are required the driers will be found the most economical course.

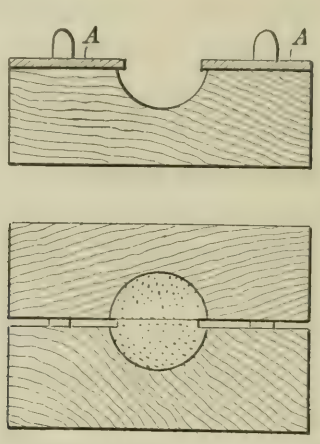


FIG. 27 CORE BOX WITH LINER FOR MAKING SELF-PASTING CORES

141 *Self-Pasting Cores.* Many intricate cores must be made in halves and then either dried on flat plates and pasted, or the two halves rocked together before pasting. If the latter is done, the binder must be of such a nature that it will form a self-pasting core. High-grade core oils when used in fairly large proportions will generally bind the two halves together in cases of this kind. In some cases, however, the expedient shown in Fig. 27 must be resorted to. One-half of the box is provided with a thin metal sheet shown at *A* which projects slightly over the edge of the box.

142 After the two half cores are made in the boxes the metal plate *A* is lifted off and the two halves of the box closed together. When the halves first meet, the condition will be as represented in the lower portion of Fig. 27. As the box is closed together, the projections of sand left by the opening in the plate *A* will crush into the faces of each half of the core so as to bond thoroughly the sand

in the two parts. This device, the writer believes, was patented some years ago by Mr. Lee, superintendent of a department of the Crane Company. In other cases the coremaker is careful to leave the sand projecting slightly above the face of the box when he strikes it off and then when the boxes are closed together there is sufficient pressure to make the sand along the faces of the core adhere. These rock-over boxes are now used in a number of different classes of work either

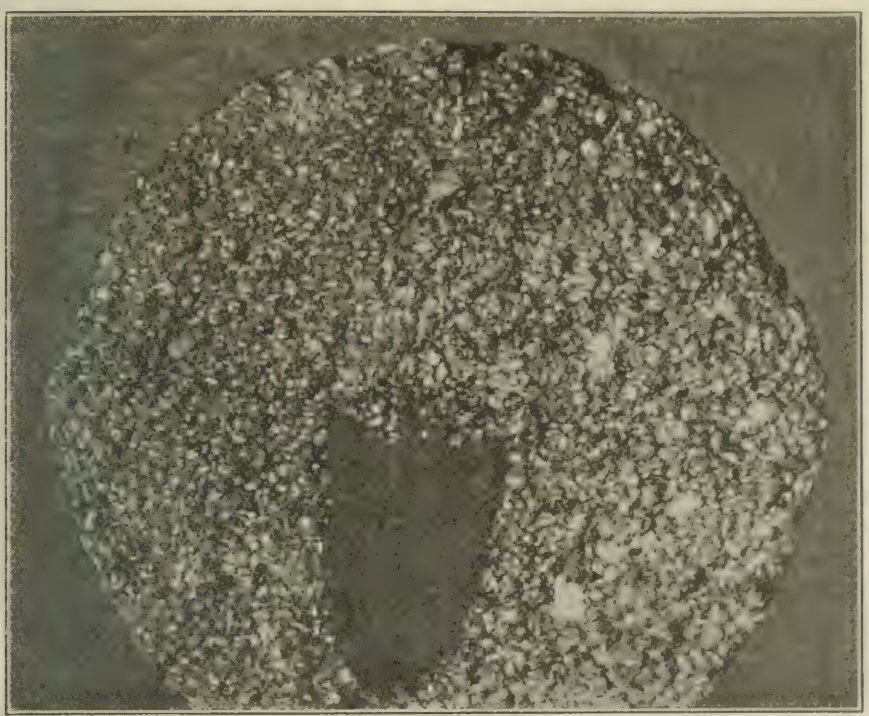


FIG. 28 MICROPHOTOGRAPH OF SELF-PASTING CORE WHICH HAS BEEN VENTED WITH A PRESS VENT

When the block for forming the press vent was forced into the lower half of the core the sand on each side of the box was forced up. When the second half of the box was placed on top of the first the sand over the vent crowded down slightly as shown and the ridges of sand at the side of the vent caused the material to blend along the parting so that the core became one homogeneous mass when baked.

with or without the stripping device and their use simplifies the making of intricate cores and at the same time permits the drying of the entire core in one piece.

143 Where cores are made in halves as shown in Fig. 27 and then rocked together, some form of block is generally employed for pressing a vent into the face of one-half of the core. The introduction of this vent block swells the sand on both sides of the vent, and this is usually sufficient to make the core self-pasting. A section of a core similarly made is shown in Fig. 28, where the use of the vent

block in the lower half has forced up two ridges of sand, causing the two halves of the core to unite.

144 *Grinding Experiments.* Concerning the compounding of core mixtures the writer has tried a great many experiments, and was aware that the grinding of molding sands greatly increased their strength, and under the impression that a similar grinding would aid core sands. In this connection a series of tests were run at the plant of the Lumen Bearing Company in Buffalo. A Wadsworth mixing and compounding mill was used, having a pair of small rollers running around on an annular track, following which are scrapers that turn the sand over first one way and then the other. A number of different classes of mixtures were put through the mill. In the case of sharp sands mixed with oil, grinding in the mill from 1 to 2 minutes was found to be a decided advantage over hand-mixing. After 5 minutes, however, there was a marked falling off of the strength of the core, due evidently to grinding the sand in such a way as to form dust which absorbed the oil and prevented its acting as a binder between the sand grains. One of the mixtures tested consisted of 120 parts of Lake sand, 60 parts of Del Ray bank sand, 5 parts by volume of No. 1 Peterson's core oil, and 1 part by volume of flour. Cores made from this mixture showed a strength of over 60 lb. per sq. in. after a grinding of 2 minutes. A large batch was made up, thoroughly mixed with the shovel, and then succeeding parts of it put through the mill, grinding the first 2 minutes, the second 5, then 15, 20, and 30 minutes. Several series of this kind were run and in each case the initial strength of the material was about 60 lb. per sq. in., while the final strength ground 30 minutes averaged less than 30 lb. per sq. in.

145 An experiment was then made by grinding a facing-sand mixture for dry sand molds. This was composed of four parts of new Zanesville molding sand, 52 parts of heap sand, $\frac{1}{4}$ part of glutrin, and $\frac{1}{2}$ part of dextrine. With this the strength increased slightly for the first 10 minutes' grinding and then remained constant for grinding up to 30 minutes. In it there is no bond that would be injured by fine material as in the case of the oil-sand cores already described. It is evident therefore, that in the case of oil-sand cores all that is required is a thorough mixing or blending of the ingredients. The hand-mixing experiments tried at the same time showed conclusively that if the oil and water were not ground together in contact with the sand so as to form an emulsion and to bring it in contact with the grains, the core would have unbonded spots.

146 The writer has also had access to the records of several foundries using mixing and compounding mills of various types and found that in all cases the oil-sand mixes require thorough blending, but are the better for not being ground with heavy rollers. Loam-sand mixtures require more grinding and in this respect are similar to molding sand. In one foundry two types of mills were in use, one a Wadsworth mill like that used in tests referred to above, at the Lumen Bearing Company, and the other one having 900-lb. crushing wheels. The latter was found to be best for making a mixture with a bonded sand and black compound. When it was first installed practically all of the old core sand had been hauled to the dump. The common practice then was to mix the sand by hand with shovels and put it through a pneumatic riddle two or three times. From 10 to 20 per cent of old sand was used in the mix. When the mill was first put in they were able to use a mix composed of the following: 25 parts of old sand, 5 parts of new molding sand, $\frac{3}{4}$ part black core compound, and 18 parts of coke breeze. This was the refuse coke cleaned out of cars in which the cupola coke was received. The coke was first shoveled into the mill, ground for between 1 and 2 minutes, and then the other material was thrown in, wet down with water and glutrin, and ground for 5 minutes.

147 After continuing this practice for a short time it was found that the sand was growing weaker. At first there was the advantage of a large amount of unburned pitch in the black compound which remained in the old core sand, but with the continued re-use of the old sand this proportion was decreased and so the following mixture was finally adopted: 20 parts of old core sand, 3 parts of new molding sand, 2 parts of sharp sand, 6 parts of refuse coke, and $1\frac{1}{2}$ parts of black core compound wet down with glutrin water at the ratio of 1 gal. glutrin to 16 gal. water. This was run through a $\frac{1}{2}$-in riddle at the mill and then used for making the largest of coarse cores. Some of it was subsequently run through a $\frac{1}{4}$-in. riddle and used for a facing sand in the large cores.

148 The selection of core materials will probably remain for some time largely in the hands of the core-room foreman, but the writer confidently expects to see all of the larger concerns and the more progressive foundries throughout the country placing the control of these supplies in the hands of the laboratory, just as they have placed the control of their metals and fuels in the hands of the laboratory.

CORE OVENS AND CORE DRYING

149 The baking or drying of a core is a complex process, depending upon the character of the sands and the nature of the binders used. Where flour, starch or dextrine are used, the moisture is first driven out and then the starch or flour compound is baked, as in any ordinary bread-baking process. To develop the greatest strength the material must not be charred, but should be carried simply to the condition of ordinary bread crust. For this purpose the temperature should be between 350 and 375 deg., and under no circumstances should it be allowed to rise above 400 to 410 deg. If flour or starch bonds are charred they immediately lose strength. Such a core taken from the oven at its condition of maximum strength will still contain some moisture in the starch or flour compounds, this moisture being chemically combined, and naturally gives off a great deal of smoke when the mold is poured.

150 In the baking of resin or pitch cores it is necessary only to heat the material to such a temperature as to cause the resin or pitch to flow and unite the grains of sand. Practically all black core compounds contain a considerable percentage of dextrine and on this account the baking partakes both of the nature of the process already described and of the melting of the pitch, causing it to flow through the sand. This requires only a temperature of 350 to 400 deg., and if heated much above 410 deg., the dextrine contained in the black compounds will be burned out or, in the case of the resin core, some of the resin oils will be distilled off and the material will lose in binding power. Most of the resin oil, however, does not distill off below 550 deg., but at 640 deg. fahr. over 90 per cent of it can be driven off.

151 In both of these classes of cores described, the first operation is to drive out the water, and the second simply a baking or melting as the case may be. In both cases, during the first part the core oven should have ample ventilation so as to drive out the steam as it is formed. During the latter part a free circulation of air is not needed and in fact not desirable, as it would tend to oxidize or burn the compounds forming the bonds if the temperature of the oven got a little too high.

152 With oil-sand cores an entirely different proposition is encountered. The active bonds are the drying oils with the exception of resin, which is sometimes blended with the core oil. The drying or paint oils all require a good circulation of air through the oven. As in the previous cases, the first action which takes place is the driving

out of the moisture contained in the core and the second largely the oxidation of the oil. If a blended or compounded oil is being used which contains mineral or tar oil, or any similar compound, there will be some volatile hydrocarbons which will be driven out. After this it is necessary to continue a good circulation of air through the oven to oxidize the oils.

153 The process going on in the core oven during the latter part of the drying of an oil-sand core is similar to that of drying oil-skin clothing. This clothing is made by dipping cloth in a mixture of oils and drying it in a current of warm air. If the rooms containing the clothing to be dried are shut up so that there is no current of air, the oil will remain moist for days, even though the temperature may be much above that ordinarily employed for the purpose. With a good circulation of air, however, the clothing will dry rapidly to a firm hard skin. In like manner oil-sand cores must be given plenty of oxygen if they are to be properly dried.

154 Very few of the bonding oils will be burned at temperatures below 500 deg., and hence when they are being used core ovens can be driven at a higher temperature than with flour or starch. There is no need, however, of carrying the temperature above 400 to 410 deg., and the writer believes that the best results are obtained at about this range. If the temperatures reach 600 deg., all forms of carbon used in core binders begin to char and the strength of any carbon bond is reduced. The cores already spoken of, composed of silica sand, Welsh mountain clay and sawdust, were baked at about 600 deg., for the purpose of charring the last ingredient. With ordinary binders, however, this would not be a proper procedure.

155 As a rule a foundryman keeps no record as to the part of his oven in which certain cores were baked and may be wholly unaware of the fact that from 10 to 30 per cent of the product is seriously damaged every day and that in many cases castings are lost on account of the cores having been baked at too high a temperature in hot sections of the oven or not thoroughly baked by being left in cold parts. With a realization of these great differences in temperature a series of experiments has been tried with a pair of recording thermometers furnished by the Bristol Company of Waterbury, Conn. Before taking up the matter of recording thermometers, however, it may be interesting to speak of one piece of bad core-oven practice which came under the writer's attention in this investigation. The ovens were simply iron boxes having bars across them to support the core plates. In the bottom of each box there was an open coke fire

with no baffle plates between the fire and the core plates. Flour cores were being made at the ratio of one part of flour to nine parts of sand, and core plates were pulled from directly over the fire with the edges of the plate at a dull red heat, showing a temperature of at least 900 deg. fahr. The cores on these plates were black on the outside and smoking as though they were on fire and were being at once put in the molds by the molders to whom they were taken and the molds poured. One of the smoking hot cores was taken from one of these lower plates and broken, showing that the outer $\frac{1}{4}$ in. was burned while the inside of it was still in the condition of

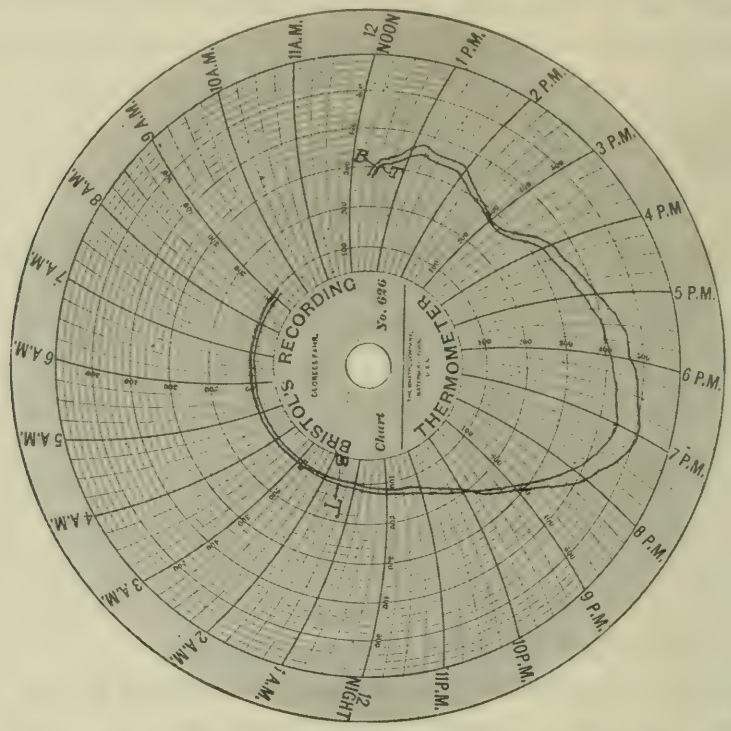


FIG. 29 DIAGRAM ILLUSTRATING DIFFERENCE IN TEMPERATURE IN A PORTABLE CORE OVEN

dough. The only wonder is that their casting loss was not greater than their report showed. The top of these ovens was probably below 550 deg. fahr.

156 The first ovens in which the writer tested the difference in temperature between the shelves was the Wadsworth oven used in experimental work at Covington and the first diagram taken is shown in Fig. 29. This heat was not used to bake cores but simply to try out some fuels and also to find the difference in temperature between the top and bottom. The curve for the lower thermometer which is placed just above the bottom shelf is marked *B*, while that

for the top thermometer which is placed just beneath the top shelf is marked *T*. In this oven the bottom shelf was the hottest while the oven was being fired. When it cooled off at night, however, the temperature of the lower thermometer fell below that of the upper. From 12.25, when the thermometers were coupled, until 2 p.m., the writer was firing with coke and trying to get the temperature up to 400 deg. At that time the coke fire was raked down and a fire of anthracite coal put in. The temperature fell from 2 to 3.10, when it commenced to rise, and at 7 p.m. the lower thermometer

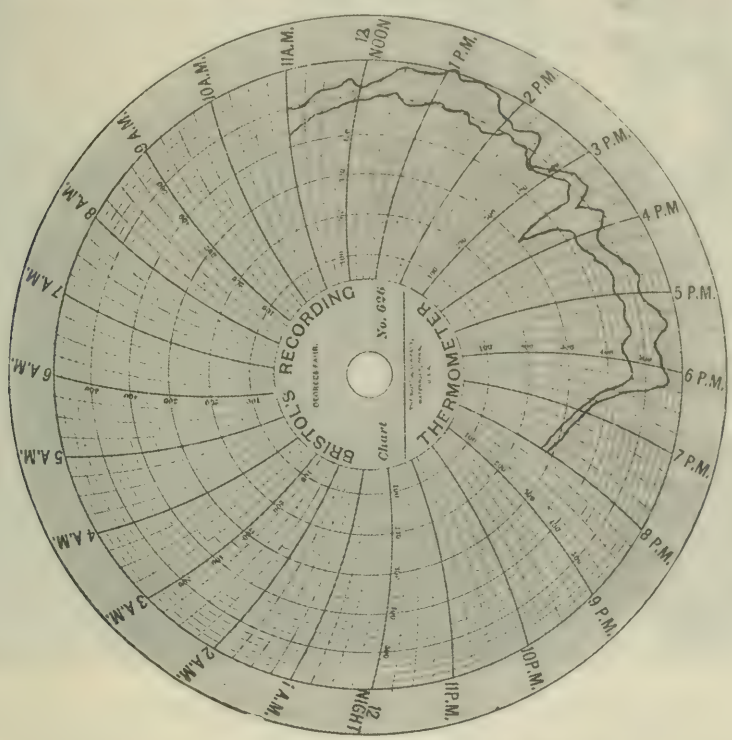


FIG. 30 DIAGRAM ILLUSTRATING DIFFERENCE IN TEMPERATURE IN AN OIL-FIRED CORE OVEN AND ALSO THE EFFECT OF OPENING DOORS AND IMPROPER CONTROL OF HEAT

had reached 510 deg. and the upper one 450 deg., there being a difference of 60 deg. between the top and the bottom of the oven. After this anthracite coal was used in this oven and the curves were kept very close to 400 deg. A difference was found between the top and bottom shelves which was rarely less then 30 deg. and under working conditions sometimes 40 deg. When the oven contained cores no difference was experienced as great as that indicated at 7 p. m. on the chart shown in Fig. 29.

157 The diagram shown in Fig. 30 was taken on a set of ovens without changing their ordinary core-room practice, the men being

allowed to open and shut doors or control the temperature as they saw fit. The ovens were of the oil-fired type and the oil came from a pipe line which also supplied a battery of oil-fired furnaces. The pressure in the line varied greatly when these furnaces were started or stopped, and unless corresponding changes were made in the control valves on the core oven, the temperatures of the core oven would increase or decrease in accordance with the varying oil pressure in the line. The oven was watched closely so that the line on the chart

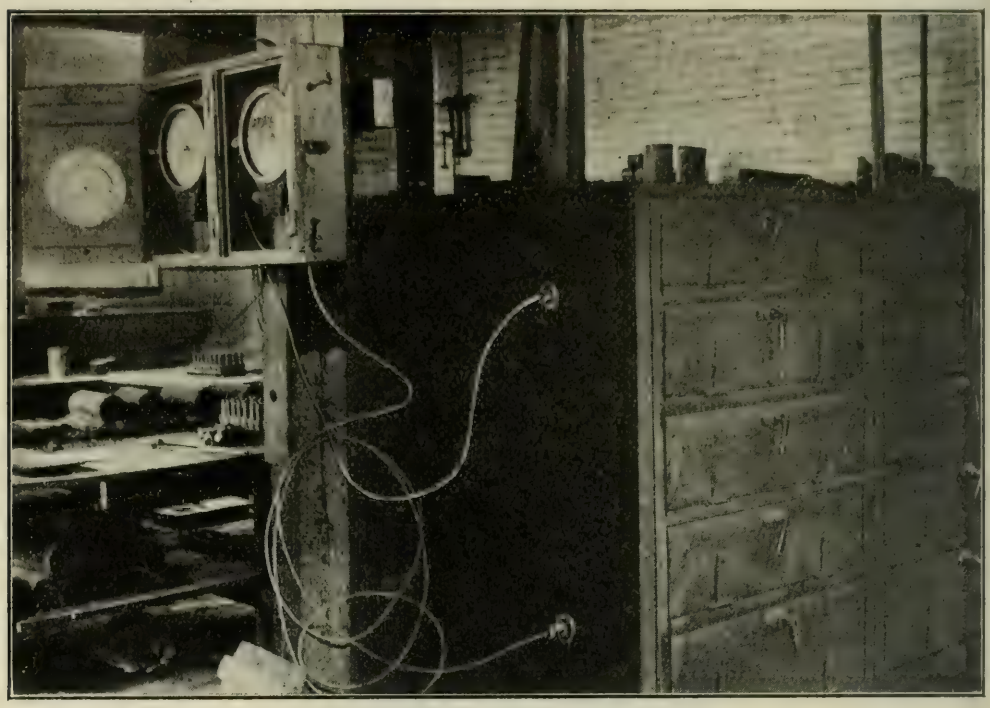


FIG. 31 OIL-FIRED CORE OVEN SHOWING MANNER OF ATTACHING THE BRISTOL RECORDING THERMOMETERS FOR ASCERTAINING DIFFERENCES IN TEMPERATURE

could be interpreted very accurately. The record for each thermometer was recorded on a separate chart but is here drawn on one chart to make comparisons more apparent (Fig. 30). In this oven the upper thermometer, the location of which is clearly shown in Fig. 31, registered the highest temperature. At 11.50 one of the men opened one of the upper doors to take out cores and left it open for several minutes. This made a rather sharp drop in the line for the upper thermometer without any corresponding drop in the record of the lower thermometer. At 12.30 a man opened one of the lower doors opposite the lower thermometer and this resulted in a sharp drop of nearly 30 deg. in the lower thermometer. At 1.10 a door located about midway between

the two thermometers was opened and both fell, the upper one showing the greatest difference of a fall of over 20 deg. while the lower one fell 15 deg. From 11 a.m. until 1.30 p.m. melting furnaces were being shut off and the oil pressure in the line was increased so that the general trend of both curves was upward, the sharp local conditions being due to the opening and closing of doors. At 1.45 two of the upper doors were opened and a lot of heavy green cores were put in. Cores were also placed on the third or middle shelf. This caused the upper thermometer to show a drop in temperature continuing until

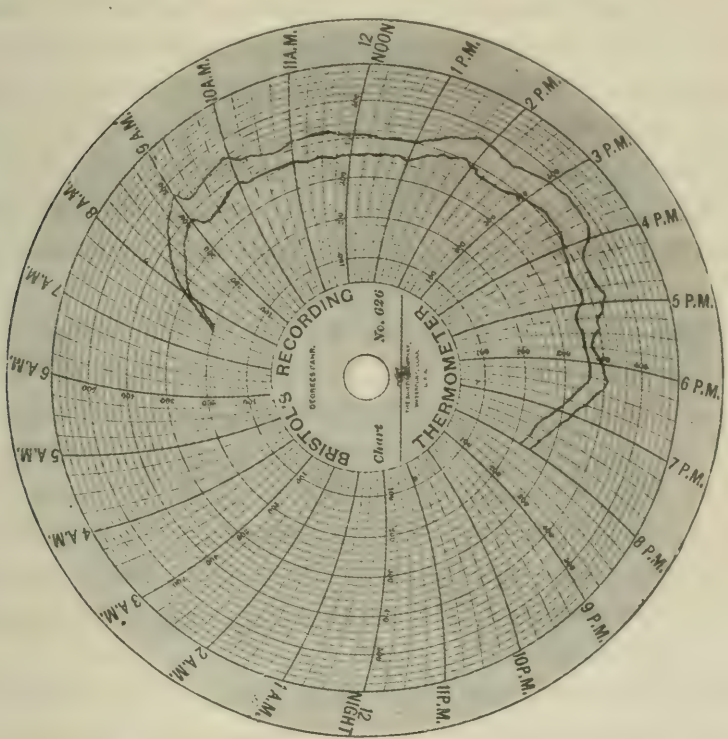


FIG. 32 DIAGRAM ILLUSTRATING DIFFERENCE IN TEMPERATURE IN AN OIL-FIRED CORE OVEN OBTAINED ON THE SECOND DAY THE THERMOMETERS WERE ATTACHED

2.10. About this time the lower door was opened giving a slight fluctuation. Just before 2.30 both upper and lower doors on the back side of the oven were opened and left open for some time and green cores put in. This resulted in a sharp drop of both thermometers. After this there was a gradual increase until 3.10 when one of the lower doors on the front side of the oven was opened and left so. The temperature fell gradually for the first few minutes and then rapidly until the lower thermometer showed a drop of over 200 deg. The

upper thermometer at the same time fell about 65 deg. In like manner the various humps of the curve can be explained throughout the day. About 4.15 the melters commenced to cut off furnaces and the oil pressure gradually rose driving the temperature up until 6.10, when the oil was shut from the oven, and the curves for both thermometers fell. On the day the chart shown in Fig. 30 was made a very large number of cores was lost on account of being burned. The next day an attempt was made to control the temperature of the oven and keep it as near 400 deg. for an average as it was possible to do. The two curves for this day have been plotted together as shown in Fig. 32. In the morning when the oven was first started the temperature was allowed to rise until the upper thermometer registered 460 and the lower one 385. The oil supply was then checked giving a quick drop in the temperature of both thermometers. The temperature picked up once more, however, until at 9.50 it was again at the same figures which it had reached at 9. Here the oil supply was again checked when the temperatures dropped slightly and continued fairly uniform until 1 p.m. Between 11.30 and 1 it was necessary to regulate the oil supply a number of times to hold the curve as uniform as it appears in this record. At 1 o'clock the writer left the oven to look after some other work and the man in charge did not regulate it as closely as he should. He did, however, check the supply at 2 p. m. and again at 3.30, thus preventing the curve rising as high as it did on the previous day. The results obtained by baking the cores at the temperatures recorded on the chart shown in Fig. 32 were much better than those obtained the previous day when the temperatures varied as shown in chart 29.

158 If all core ovens were equipped with recording thermometers and account taken of the time different cores were placed in the oven, the foundryman would have a record of the baking temperature of each batch of cores and would thus be able to trace irregularities in the quality of the cores to the baking temperatures, if variations in these temperatures were to blame. When workmen know that a recording instrument is watching them and drawing a record of just how they maintain the temperature in the oven they will look after the fire-and the drafts and see that the record is a credit to them. The writer believes it is a good thing to put the recording thermometers where the foreman can see them and then hold him responsible for the results, for many castings are ruined through bad cores and the trouble laid at the molder's door or considered a mysterious dispensation of Providence.

159 *Core-Oven Design.* It is not intended in this paper to go into an exhaustive discussion of core-oven construction, but the underlying principles of core-oven design will be stated. If it is to be efficient it must be so constructed as to maintain a practically uniform temperature throughout the entire drying chamber. For the best results the extreme variations between different parts of this chamber should not exceed 60 deg. fahr. There also should be a proper circulation through the oven to insure the carrying off of any steam generated during the first period of drying of cores and to afford an ample amount of oxygen for oxidizing and drying oil binders. Where no oil binders are used and the cores are dried in batches it is advisable to vary the amount of circulation through the oven at different periods of the drying process so as to avoid undue oxidation of certain classes of binders toward the end of the drying period.

160 The most common mistake made in core-oven design is that of locating the firebox too close to the oven and of discharging the products of combustion into the core-drying chamber at too high a temperature. This practice renders certain portions of the oven inoperative for efficient baking on account of the fact that if any cores were located in these hot parts they would be burned. In the oven there should be ample space between the combustion chamber, or firebox, and the oven, or drying chamber, for the mixing of the products of combustion so that they will have a uniform temperature when entering the oven, and this should be the maximum baking temperature of the oven. The most efficient ovens that have come under my attention have all had their firebox located at some distance from the oven proper, or else have had a large mixing chamber introduced in the flue between the oven and the firebox. In one case there is a battery of ovens constructed with fireboxes in the basement and the ovens are located two stories above. The flues are so arranged and provided with dampers as to make it possible to control the temperature and volume of the incoming gases entering each oven or oven compartment. A provision of this kind insures rapid drying without any danger of burning the cores.

161 *Steam-Heated Core Ovens.* For some classes of binders steam-heated ovens are excellent since they insure control of the temperature and preclude any possibility of its rising above the desired maximum. The fuel economy of the steam-heated oven, however, is very low, and hence some form of direct firing by carbonaceous fuel is generally depended upon.

162 *Core-Oven Fuels.* The fuel used must be one that does not give a smoky flame which would cover the cores with soot. In consequence, anthracite coal, coke, fuel oil, and gas, are the fuels commonly employed. The products of combustion from anthracite coal and coke are carbon dioxide with possibly a little carbon monoxide if there is imperfect combustion. With these are mingled a large amount of nitrogen which is passed through the oven with the air supply to provide oxygen for the combustion of the solid carbon. All of these gaseous products are efficient drying agents and have the power of absorbing moisture and carrying it from the core oven with them.

163 Turning to fuel oil or gas as a means of heating the oven, an entirely different set of fuels is encountered. Both of these fuels contain larger percentages of hydrocarbons, all of which burns to water, which passes off as steam and as its temperature falls in the core oven it tends to saturate the air with steam and hence is not efficient as a means of carrying moisture from the oven. In other words, while all of the combustible material in oil or gas produces heat when it is first burned, the products of combustion are of such a nature that much of the heat is carried through the oven locked up in the steam resulting from the burning of the hydrocarbon compounds and so is not available for the drying of cores. The exceedingly high thermal efficiency of gas and oil, however, and the fact that the use of fuels of this kind does away with the handling of coal or ash may make it advisable to use such material though it is not the most efficient fuel.

164 *Continuous Core Ovens.* In the case of continuous ovens where the cores are carried by mechanical means, care should be taken to see that there is no hot spot in the ovens where the temperature exceeds the safe drying limit for the binder in use. Cores may be made to resist high temperatures by spraying the surface of the cores to increase the percentage of binder in the surface, but this is expensive since in spraying cores much of the material is wasted, and the operation itself takes time. It is far more economical to design the oven so that it will bake the cores with the maximum degree of rapidity and at the proper temperature, than to try to utilize improperly designed ovens by doctoring the core-room practice. One advantage of continuous ovens over many of the other types is that the temperatures of the different parts may be controlled so as to give the most rapid and efficient drying possible, but this is true only where the cores are approximately uniform.

165 *Chamber Ovens with Shelves.* For chamber ovens in continuous use, the cores being dried on individual shelves, it is necessary to determine the condition of the drying by inspecting the cores. In such ovens the products of combustion usually rise and pass out of the top and if there are partially baked cores on the upper shelves and subsequently green cores beneath them, the steam driven out of the green cores passes over the dry cores and the moisture may, to some extent, be absorbed by the dry cores, thus retarding their baking. In any case the presence of steam in the ascending current of air will retard the baking on the upper shelf. This is one reason why the lower shelves of a drying oven frequently bake faster than the upper shelves, even when the temperature of the incoming gases is kept well below the maximum allowable for the oven.

166 *Distortion of Metal Parts of Core Ovens.* Core ovens have been arranged in many cases so that the fire was 1 to 3 ft. from the cores being dried. The core cars have often been warped and bent out of shape by the fact that the flame from the fireboxes played against them. Iron will not be distorted at any temperature that is safe for the baking of cores, hence any warping of any metal work in the oven is evidence of too high temperature. For the proper regulation of the temperatures in core ovens, recording thermometers should be applied not only to the ovens themselves but to the main heat flues.

167 *Core-Oven Cars.* For efficient core-oven work the car must be designed so that it will support the cores without any danger of being distorted. The tracks for supporting the car must also be rigidly supported and kept in perfect condition. Too little attention is given to the design of this part of the core-room rigging. A firm installing a core oven will frequently order from the lowest bidder. From the day such an oven comes into commission it is a constant expense in the way of spoiled cores and time expended in unnecessary manipulation in getting the work to and from the oven. The core trucks or cars should run into the oven as easily as any other part of the industrial transportation system, and there is no need of having to bring a 25-ton travelling crane to the front of the core oven and couple it to the truck by an intricate system of chains and blocks in order to shove 5 tons of cores into the oven.

CORE PASTING, HANDLING, AND STORAGE

168 With the increasing demands from the designing department for greater accuracy in finished castings, it has been necessary to

improve the core-room rigging to insure the accurate fit and location of the cores. For convenience many cores are made in halves so that they can be dried on flat plates. This method also affords an opportunity for the forming of vent passages on the parting. Fig. 33 shows a pair of half cores and the vent block which was used in pressing vents in the cores. The horizontal strips attached to the blocks press grooves in the face of the cores that lead the vent to the face of the print and the vertical or projecting pins press vents into

FIG. 33 BLOCK FOR FORMING PRESS VENTS IN CORES AND PRESS-VENTED CORES

the pockets of the core. After baking the half cores must be assembled and pasted together.

169 *Core-Pasting Device.* John Gow of the General Electric Company has contrived and patented the device for core-pasting shown in Fig. 34. It consists of a shallow wooden box made of pine, well protected with shellac. Through the bottom of this box is cut a series of grooves corresponding to the line of paste required to unite the parts. The core shown at the left is laid in the box, being guided into place by the three guide blocks. The box with the core in place is then placed in a tank containing 2 or 3 in. of liquid paste. The box

is pressed into the paste so as to force it up through the groove into contact with the face of the core. When the box is lifted out once more and the core removed there will be long narrow strips of paste where they are required on the face of the core. The two halves of the core are then assembled and placed in the drying oven to dry the paste. This method is economical of paste and applies the material very much more rapidly than it could be done by hand. It also

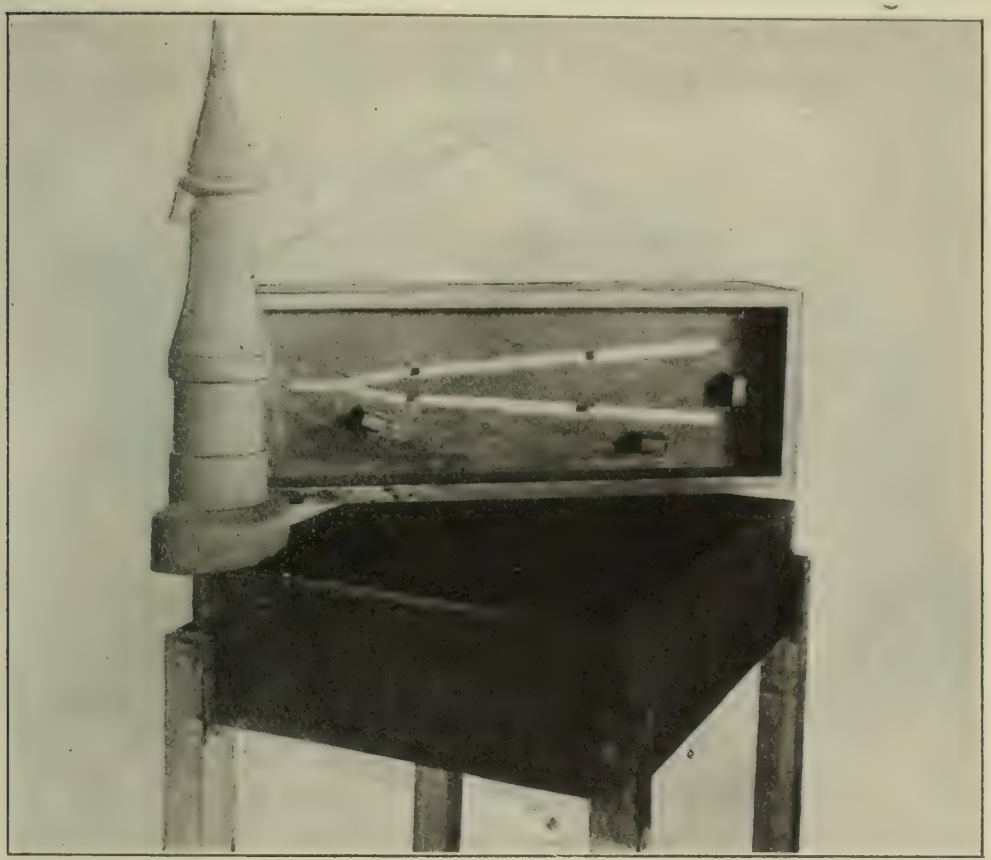


FIG. 34 DEVICE FOR PASTING SPLIT CORES

avoids all possibility of getting paste into the vents. Where duplicate work is to be done it has proved an excellent time saver.

170 *Drying Pasted Cores.* After the cores are pasted it is necessary to dry the paste before introducing the cores into the molds. In some cases a torch is used to dry the paste near the edges of the parting and the cores sent directly to the mold, but the maximum adhesion of the paste has not been developed and the core is likely to shift. Generally a regular core oven is used for drying, the pasted cores being placed on plates and introduced on to core-oven cars or placed in drawer ovens.

171 A device for pasting and drying cores in large quantities is shown in Fig. 35. The table supporting the core plates consist of angle iron legs which carry angle iron rails running along the sides. These in turn support cross bars carrying a pair of ordinary railroad rails upon which the core plates may be placed. The hood is lifted off while the cores are being assembled. The cores are pasted on the plates with the use of suitable portable pasting jigs. As soon as one entire bench of cores has been pasted the hood C is dropped over

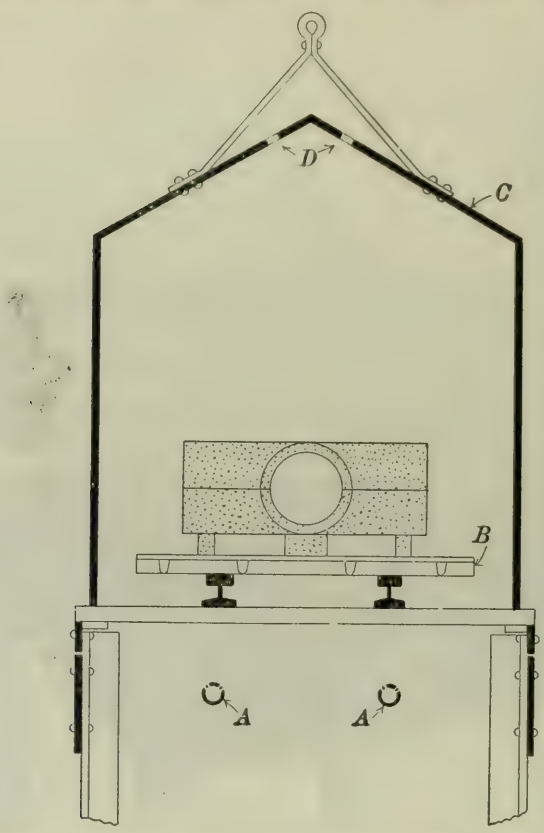


FIG. 35 PORTABLE HOOD USED IN DRYING PASTED CORES

the structure as shown. The gas pipes AA are provided with perforations through their entire length and when C is dropped in position the gas is turned into the pipes and lighted. The heat from the gas jets issuing from the pipes AA rising under the core plates B is deflected by them to the sides of the hood, and surrounds the cores with hot air so as effectually to dry the paste. For ventilation a series of small holes near the top of C is arranged as shown at D. As soon as one bench full of cores is covered with its hood and the cores are drying the workmen pass to the next bench and begin to paste cores on it. By the time they have pasted them on four or five benches

those on the first are dried, so that the gas can be turned off, the hood lifted and the cores removed. This scheme does away with the handling of the cores while the paste is green and avoids possibilities of cores shifting.

172 *Core-Assembling Jigs.* For assembling automobile cores various types of jigs have been devised. The principle on which these jigs work is well illustrated in Figs. 36 to 39. Fig. 36 represents a jig for filling the pasting faces of cores. The upper edges of this jig are protected with pieces of tool steel which have been made as hard

FIG. 36 JIG FOR FILLING PASTING SURFACE OF JACKET CORES

as fire and water could make them, so that they will not be affected by passing a file over them. The core is dropped into the jig shown in Fig. 36 and a coarse file passed over the pasting face so as to bring it to a uniform surface. The core boxes are so constructed as to allow about 0.01 in. on the core face for filing. The lower half of the core is then placed in a jig as shown in Fig. 37. The paste is applied to the surface and the next section of the core placed in position as shown in Fig. 38. In this case it was a small section at the back of the core. The paste is then applied to the next surface and the upper half of the core is introduced as shown in Fig. 39. In taking the illustration a set of unfiled cores were assembled in the jig so that the parting is not perfect, as it would be in the case of filed cores. After the cores have been assembled in the jig they may be removed and placed on

the core plate for drying in the oven or the entire jig may be placed in the oven and the cores dried in the jig. Where a limited number of

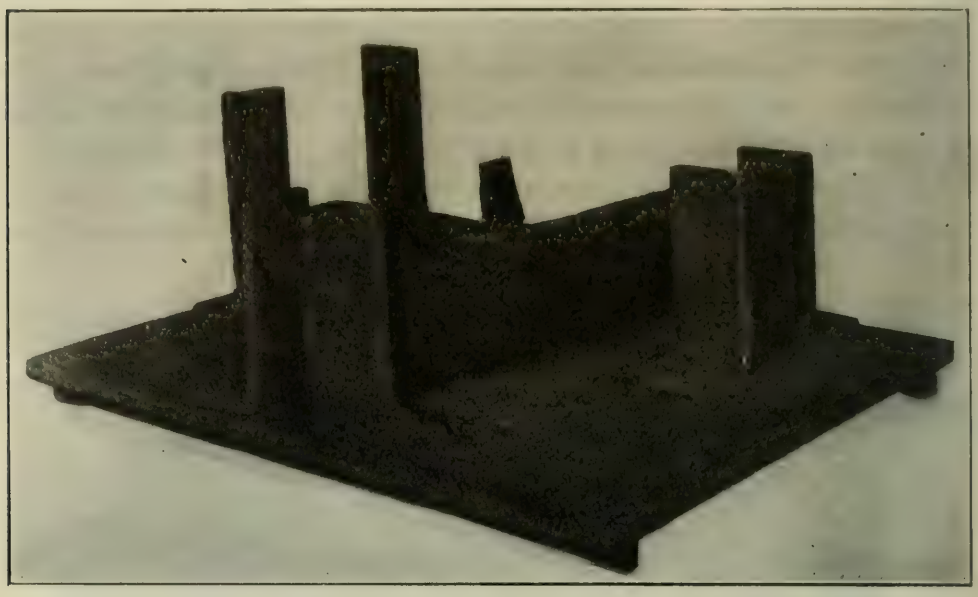


FIG. 37 FIRST SECTION OF CORE PLACED IN PASTING JIG

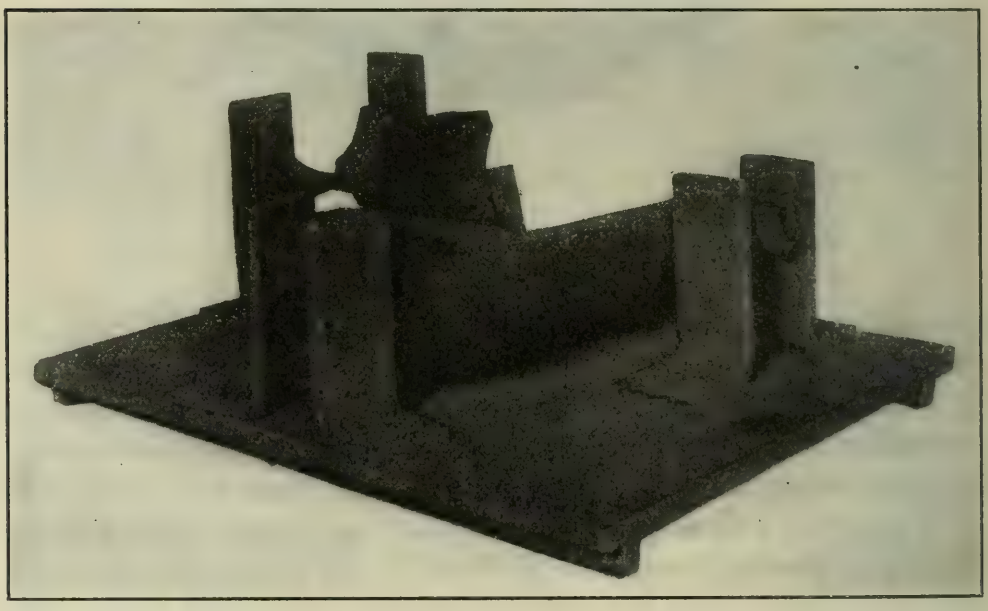


FIG. 38 PASTING JIG SHOWING SECOND SECTION OF CORE IN PLACE

cores is required it is best to dry the cores in the jig, but with a large number the amount of equipment in the shape of jigs would be prohibitive, and hence the cores are removed before drying.

173 Where a large output of cores is required these jigs are made entirely of metal, and it is no uncommon thing for the modern foundry-

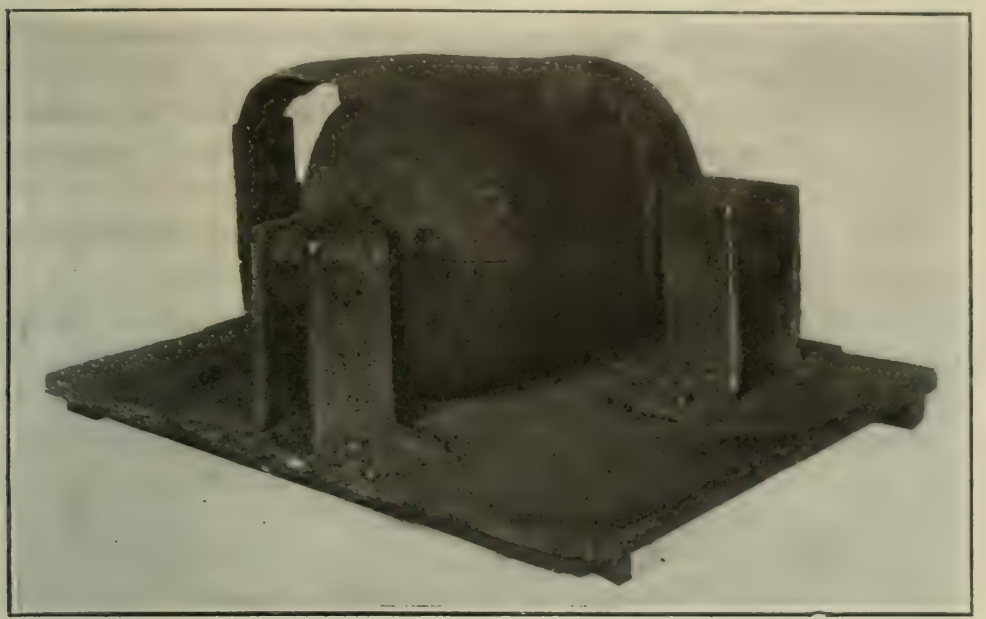


FIG. 39 PASTING JIG SHOWING COMPLETED CORE IN PLACE

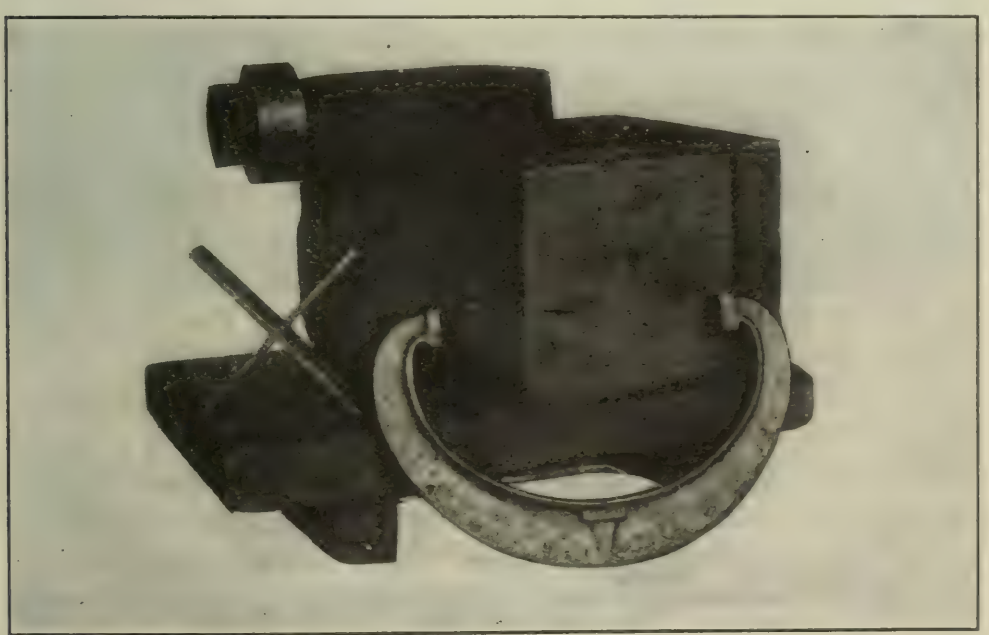


FIG. 40 GAGES FOR TESTING CORE

man to hold his core work within limits of a few thousandths of an inch for medium sized cores and within 0.01 in. for cores 1 ft. or more in diameter or length.

174 The use of core jigs is not confined to small work such as cores in connection with automobile castings, but may be applied to steam engines or any large work. A wooden or metal frame may be constructed with bearing points corresponding to the bearing points which carry the cores in the mold. The cores to be tested can be lowered into one of these jigs with the crane and then by means of suitable gages the metal thickness that will be left at all points can be ascertained before the core is placed in the mold.

175 *Core-Testing Gages.* After the cores have been dried they are tested by means of various gages, a set of which is shown in Fig. 40. The core is placed on a steel plate and the shoulder at the right is filed off by passing a file over the hardened steel plates surrounding this portion of the core. This insures the proper length of the core from shoulder to shoulder. The interior diameter of the core is tested with the star gage shown at the left; the exterior diameter with the caliper gage shown at the right; and the print by the ring gage shown at the upper left-hand portion of the illustration.

176 *Core-Setting Jigs and Gages.* It is also possible to use setting jigs for placing cores in the mold to insure the proper fit of the various parts. Fig. 41 represents the drag half of a cylinder mold for a large gas engine. In this case the cores are supported on arbors which fit in prints at the ends of the molds, the prints being shown at the front and back of the mold. Fig. 42 illustrates the same mold with the main barrel cores in place and the core jig located over the top of the barrel cores to measure the setting of the barrel cores. This jig may be placed anywhere along the length to see if they are straight and properly set at all points. The illustration shows a type of cylinder for a 2-cycle engine, the compression cylinder being at the right and the ordinary working cylinder at the left. The compression cylinder does not need water jacketing.

177 The drag half of the mold with all of the cores assembled is shown in Fig. 43. After the upper half of the jacket core is in place a caliper gage may be used to see that the space between the barrel and the jacket is correct and a gage similar to that shown in Fig. 42 may be used to test the outside of the jacket core if desired.

178 *Handling the Dry Cores.* The handling of cores is of importance. Every time a core must be picked up and set down there is danger of its being broken and an expense for the labor involved. Cores should pass from the oven to the molder with the fewest handlings consistent with the maintenance of a sufficient stock of cores to

insure continuous work in the foundry and also with distribution of the work in the core room so as to allow the coremakers to complete batches of a given style of cores in the most efficient manner.

179 Where the operations are such that a constant number of standard cores are required they should be taken from the core-oven trucks, and placed directly on some form of carrier which will deliver them to the molders. In the case of small chunky cores such as are used in fitting shops the cores are often piled into boxes and these in

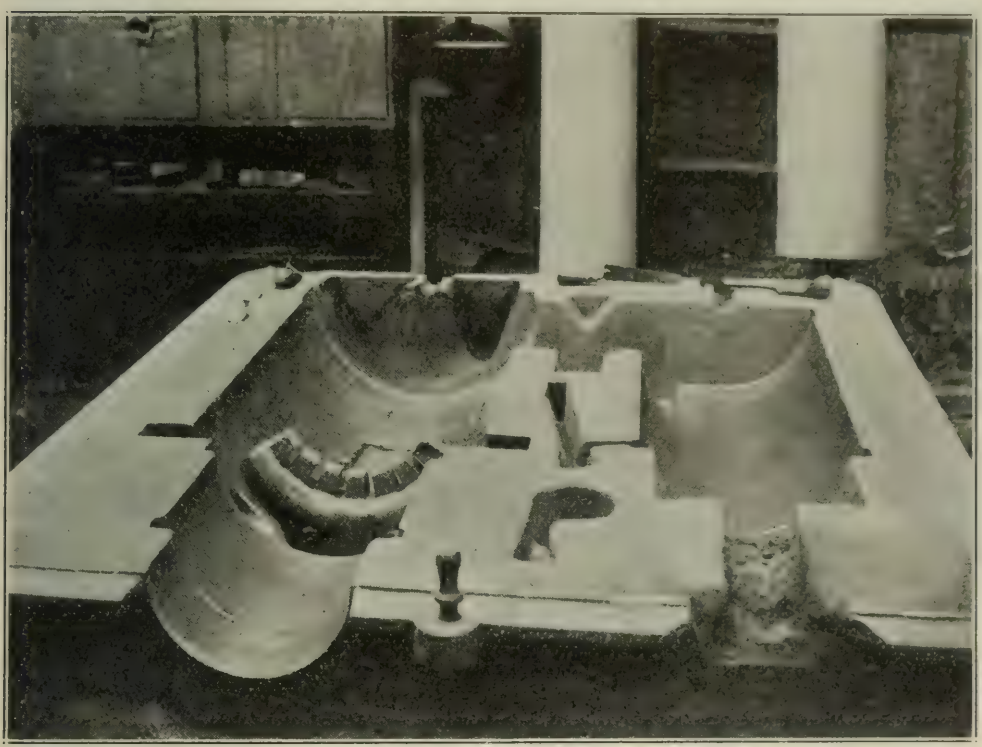


FIG. 41 MOLD FOR LARGE 2-CYCLE GAS-ENGINE CYLINDER WITH PORT CORES IN PLACE, SHOWING SUPPORTERS FOR CORE ARBORS

turn laid on the trucks or on racks supported from overhead trolleys and then transported to the molders. Where cores must be passed to a storage or must be pasted subsequently to drying, they can be taken from the core-oven trucks, laid on industrial railway cars and delivered to the storage or pasting departments and to the foundry as required.

180 *Handling Green Cores.* Where possible the coremaker should place the plate carrying the green cores either on the shelves of the core-oven truck in which it is to be dried or on a conveyor which will carry the cores to the core-oven. For medium sized plants one

of the most efficient methods is to have the core-oven trucks so designed that they will run on tracks between the coremaker's benches. The coremakers place the cores on the shelves of the truck and laborers then run it out from between the benches to a transfer car which delivers it to the oven in which the cores are to be baked. After the cores are dried the truck carrying the dry cores may be taken to the core storage or the dry cores may be placed on

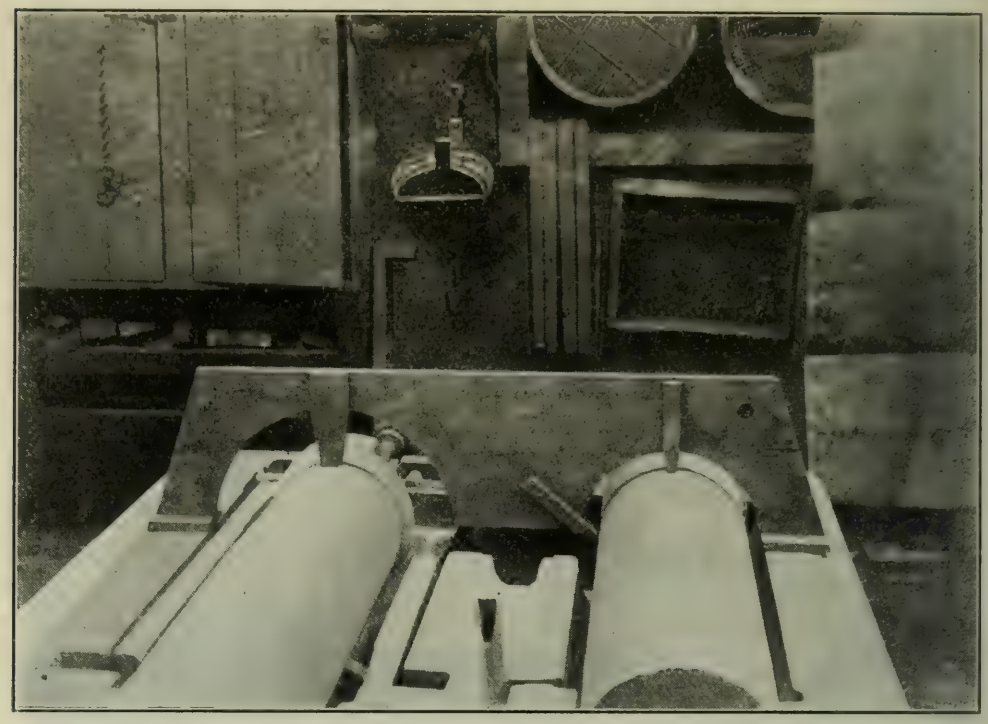


FIG. 42 MOLD FOR 2-CYCLE GAS-ENGINE SHOWING LOWER HALF OF JACKET, CORES AND BARREL CORES IN PLACE, WITH GAGE FOR TESTING CORE SETTING

industrial railway trucks or carriages supported from a trolley system and carried either to the storage or the foundry.

181 *Protecting Cores from Dampness.* Complaints are constantly made to the effect that flour, starch, dextrine, or glutrin cores become soft if left in storage. The reason for this is that if exposed to moisture-ladened air these cores will absorb moisture. The author has seen a core storage located in a cold unheated building with the exhaust from an engine playing past the unclosed windows so that the exhaust steam came in over the cores in clouds. They might just as well have played water from a hose over them so far as the effect was concerned.

182 On the other hand, there are a good many heated core storages. In most cases these are located over the core ovens and take advantage of the waste heat from the core ovens. In other cases they are located over malleable annealing ovens or in any place where there is waste heat available. In some other cases they are heated with steam coils or stoves fired with coke or natural gas. The author has seen flour cores taken from such a heated storage and used after they had been in storage for three years, and they were

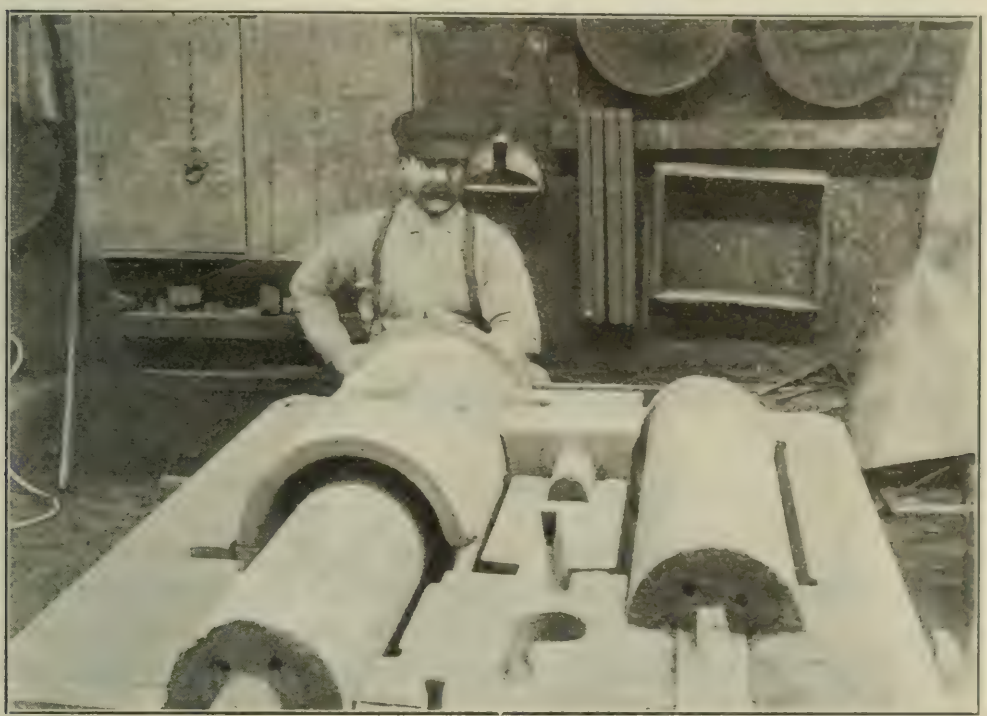


FIG. 43 MOLD FOR 2-CYCLE GAS-ENGINE CYLINDER, CORES SET

just as good as the day they were made. Glutrin cores made with gangway sand have been used after they were many months old.

183 In many plants it is the practice to leave cores standing upon the molders' floor from one to three days before they are used or to introduce them into molds long before they are used.

184 *Moisture in Molding Sand.* In order to ascertain the moisture conditions that the core must meet in the mold, samples of the molding sand were taken from many foundries in different parts of the country and moisture determinations made. The amount of moisture used in molding sand was found to vary from less than 5 per cent to over 10 per cent, and in the same foundry it will frequently vary 3 per cent in a single day. By watching the molding operations

the author has been able to predict which molds would be lost before they were poured, and these predictions were verified when a badly blown casting was shaken out. In one case the foundryman had been blaming first the iron and then the coremaker, it was proved conclusively that the trouble lay in the too free use of the swab on certain molds. A core made from any of the standard binders should be able to remain in the mold from morning until afternoon without any danger of its absorbing an excessive amount of moisture, providing the core has not been subjected to undue moisture in the storage or the molding sand and has not been worked too wet or overswabbed.

185 *Location of Core Storage.* The core storage in any plant should be so designed and located that the cores can be handled with the least amount of labor and also so that the number of cores of any given kind on hand may be ascertained readily at any time. The placing of the core storage on the second floor usually makes it possible to utilize some waste heat for the protection of the cores, and the transportation of the cores to and from the storage by elevator adds but little to the expense, as vertical transportation is very cheap.

CORE MACHINES AND CORE RIGGING

186 *Types of Core Machines.* Core machines in so far as they affect core sands and core-binding materials are considered in this paper. There are four distinct types of core machines in general use. The best known and probably the oldest type is that which corresponds to the hand-rammed molding machine. In this case the core boxes are simply attached to a molding machine, the core sand rammed in the boxes and the machine used for rolling over and drawing the boxes away from the finished core.

187 Another type of machine which has long been in use is the screw-feed machine forming a core through a die. The plunger-feed machine works on the same principle so far as the die is concerned. In a machine of this kind the core mixtures must be forced through a die and on to a plate. Such a mixture must contain sufficient green binder to make the core stand up on the plate and sufficient oil to make it slip on the plate without being torn up. These conditions are antagonistic, as the green binder depended upon is usually flour and the core oil neutralizes much of the effect of the flour. Core mixtures for use in machines in which the core is forced through a die must contain an excess of binder, but the capacity of the machines is usu-

ally sufficient to more than pay for the excess of binder used. Such cores usually vent fairly freely in spite of the excessive amount of binder and the reason for this is that the large amount of flour used shrinks greatly during drying, thus giving added vent space.

188 Where silica sand or a good sharp sand is used as a base with a proper binder the cores will neither shrink nor swell appreciably during baking, and in fact, George H. Wadsworth has succeeded in producing both square and round cores which were kept within limits of 0.004 in. over or under a given size. This is a total allowable variation of less than 0.01 in. Machines of this type, that is, either the screw-feed or plunger-feed machine, are extensively used for forming parallel sided stock cores.

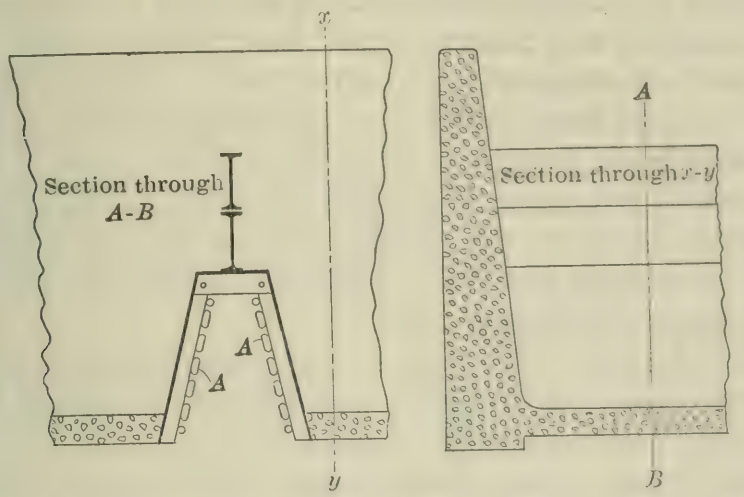


FIG. 44 SECTIONS OF BIN FOR STORING LARGE QUANTITIES OF SAND

189 For forming irregular cores a new type of machine has come on the market within the last few years. This is known as the Hulet machine, manufactured at Kankakee, Ill. In this the core mixture is blown into the core box by compressed air and the packing accomplished by the impact of the sand as it enters.

190 The jar-ramming molding machine was very promptly applied to core making, and a large variety of special jar-ramming machines have been developed for this purpose. Jar-rammed cores vent more freely than hand-rammed cores on account of the fact that hand ramming tends to form hard faces which effectually block the flow of the gases through the mold and so interfere with the vent. In the case of automobile cylinder jacket cores and other intricate work made from silica sand and oil mixtures jar ramming has almost eliminated the use of artificial vents.

191 In the case of large cores made from pitch or black compound mixtures or from any mixtures suitable for heavy work the jar-ramming machine can be used with special or ordinary standard boxes. A layer of facing sand is first shoveled in next the core box, the core arbors, and the core bars are introduced together with any coke or other material that may be required to form vent passages, the remainder of the sand is then introduced and the whole mass jar-rammed, the core arbors and core bars settling down into the sand as it is rammed. This results in a uniformly rammed core which does not have a tendency to swell and become irregular in drying as is often the case with hand-rammed cores. Such a core also vents more freely than a hand-rammed core.

192 These advantages alone would be sufficient to induce foundrymen to introduce core machines into their foundries, but there is the additional advantage of larger output and the author is convinced that in the future there will be an increasing use of machines for forming cores in the foundry. In addition to the types of machine already specially referred to nearly every type of molding machine has found its application in the making of some kind of special cores. The roll-over molding machine is being used extensively for the production of automobile jacket cores.

193 *Sand-Handling Equipment.* Particular attention should be given to the storage, preparation and mixing of core sands. The sand should be put into storage at the time of year when it is driest, as this saves freight and also insures sand in better condition for core work. Core sand should always be kept under shelter.

194 The method of storing sand will depend largely upon the size of the plant. For comparatively small plants a series of covered bins may be arranged in the basement and the sand introduced into them through chutes in the side of the building from a switch outside or dumped into them through hatches in the roof.

195 For the storage of large quantities of sand, concrete bins are best. Fig. 44 illustrates the bin construction which has been used in a large steel foundry. The bins are separated by division walls, the lower part of which are made of steel plates supported on angle irons as shown. Inside of these low walls are steam coils *AA*, through which steam is circulated so as to dry and thaw out any excessively damp or frozen sand which may be received in bad weather. The upper part of the wall between the bins is simply composed of I-beams bolted one on top of the other. In a case of this kind the sand can be handled into the bins and from there to the foundry by means of a grab bucket operated by means of an electric crane.

196 For smaller plants some conveyor system is advantageous, the sand being conveyed from the bins to the mixing machinery by means of a belt conveyor. Where the cores are made by hand the core sand can be delivered to the operators' benches either by means of boxes carried on trucks or supported from overhead trolleys, or by a conveyor system, but in most cases the latter is more complicated, and for this reason not desirable. Another objection to it is that in most plants several core mixtures are in use and in some cases the same operator may work on different mixtures during different parts of the day. For moderate sized plants working only from four to ten bench coremakers, a mill of the Wadsworth type, that is, a small mixing and compounding mill with small rollers gives excellent results. The centrifugal core-mixing machine made by William Sellers & Company, Inc., of Philadelphia, has a larger capacity and is used in many plants.

197 The experiments tried in the mixing of sands at different foundries indicate that for all sharp sand mixtures the sands should be thoroughly mixed but not ground, and for this reason a paddle mixer working on the principle of the ordinary pug mill used in brick manufacture is suitable. The Standard Sand and Machine Company of Cleveland, Ohio, were pioneers in the making of this class of machinery and have developed combination paddle mixing and grinding machines suitable for various core mixtures and various capacities. In any case the plant should be equipped with some kind of mixing machinery, for hand mixing is not only expensive, but is not capable of uniform production and hence an excessive amount of binder would be used.

NOTE OF ACKNOWLEDGMENT

The apparatus comprising the equipment of the laboratory at Covington, Va., was furnished by the following firms: Wadsworth Core Machine and Equipment Co., of Akron, O. The Acme Core Machine Co., of Cuyahoga Falls, O.; Detroit Core Machine Co., Detroit, Mich.; The Robeson Process Co., Covington, Va.; Bristol Co., Waterbury, Conn.; W. S. Tyler Co., Cleveland, O. The lenses were loaned by Spencer Lens Co., Buffalo, N. Y., and Sauveur and Boylston, Boston, Mass.

To test the advantage of milling core sands the Lumen Bearing Company of Buffalo, N. Y., placed at the author's disposal their Wardsworth compounding mill and core-room equipment. The author is also indebted to the Detroit Testing Laboratory for assistance in the examination of sands and advice in regard to methods, to Prof. Geo. W. Cavanaugh for work done on sands at Cornell University, and to many foundrymen throughout the country who have assisted in the work by furnishing information concerning their experiments or by carrying on experiments at his suggestion.

The micrographic investigation into the structure of cores was one of the most difficult phases of the work which was taken up. In this work assistance was received from the Detroit Testing Laboratory, of Detroit, Mich.; the Spencer Lens Co., of Buffalo, N. Y., and from Sauveur & Boylston, of Boston, Mass.

In connection with the investigation into the colloidal properties of sands the author wishes to express his indebtedness to George K. Elliott, of the Lunkenheimer Co., of Cincinnati, Ohio, to Prof. Edward Orton, of the Ohio State University, and to the Agricultural Department and the Geological Survey at Washington.

The following manufacturers of foundry supplies have furnished standard core compounds or have given valuable suggestions and hints for the conducting of the work: The National Core Oil Co., Buffalo, N. Y., Spencer Kellog and Sons, Buffalo, N. Y., Frank L. Young & Kimball, Boston, Mass., The Cutter Wood Supply Co., Boston, Mass., The Hill & Griffeth Co., Cincinnati, Ohio, The J. D. Smith Co., Cleveland, Ohio, and The C. E. Mills Oil Co., Syracuse, N. Y. A number of chemists and metallurgists in different parts of the country have also coöperated by supplying information or advising as to the methods of procedure. Among these are William M. Davis, Lubrication Engineer, Boston, Mass., Alex. E. Outerbridge, of William Sellers & Co., Philadelphia, Pa., N. W. Shedd, of the Buffalo Foundry Co., Buffalo, N. Y.; also the chemists in connection with the Forest Testing Laboratories, of the United States Forest Service, at Madison, Wis.

DISCUSSION

B. D. Fuller, in a written contribution, agreed with Mr. Lane that the machine shop ought to demand as perfect a finish on the interior of the casting as on the exterior. It is only a question of method, equipment and materials. Many core rooms are hampered by prejudice as well as poor equipment. As to the method of mixing sand, working the mass over by hand until the binder seems to be thoroughly distributed throughout the sand, adding water, and mixing it by rubbing with an iron pipe, is probably efficient where a small quantity is to be mixed, as was the case in the experiment. The rubbing or mixing with the pipe is somewhat similar to the action of the millstone in the grinding pan; it smashes the grain and destroys the vent channel by filling it with fine material. A better method is to cut over and rub the grains one against the other, using mixing machines which mix after this fashion. Mechanical mixers will save from 25 to 50 per cent of binder over hand mixing. For the transportation of cores, when the output consists of a specialty which insures uniformity to a degree, trolley or carrier transportation is efficient and economical. Where cores are large, the traveling crane, intershop tracks with easy running trucks, or electric locomotives, should be used. Stiff cast plates should be used capable of

carrying a load without bending or buckling and cores marked and loaded for distribution by a capable man at the bench.

As to the sand for cores, Mr. Fuller stated that the strongest cores he had been able to produce were made from crushed and well washed silica containing no dirt and grain of uniform size. He is, however, opposed to the abolition of venting in cores, as referred to by the author, when made of silica, sand and oil, although successful cores are so made at times.

As to the oil for bonding cores, linseed is the best, chinawood, soya bean and corn are also good. A bond which sweats to the surface will give trouble. Molasses and glutrin will do this, but, if everything be just right, either will make good cores when used in conjunction with oils and silica sand. Imperfect mixing and drying causes, however, much loss, and places the balance in favor of the use of oil. In large cores, cast-iron arbors which give rigidity will be found of assistance; they are more efficient and cost less than the wrought-iron bars generally used.

As to the cost of a cubic foot of core, in particular as far as it depends on the number of cores which a man can make per day from a given mixture, Mr. Fuller pointed out that a mixture of fine silica from the region of Zanesville, Ohio, together with a coarser silica from another part of the State, being used with oil, enabled the coremaker to make at least one-quarter more cores per day than when a pure white silica, known as "glass sand," and oil was used. The glass sand hangs to the boxes to such an extent as to require constant cleaning of boxes, taking much time.

Another marked difference in these two mixtures is that the glass sand is so clean and open that the oil and water used will not stay in suspension in the sand, but will settle, giving dry sand at the top and soggy, wet sand at the bottom. The Zanesville silica, however, contains enough natural binder to overcome this tendency. The glass sand gives much better results when flour or dextrin are used as a binder in place of oil.

An excellent design for ovens for drying large cores or molds is one where the fire-box is in a pit, and the heat is conducted under the floor of the oven from the rear to the front, then up into the oven and back through the cores or molds to the flues, placed in the rear of the oven. A mistake is often made here in not building the fire-box sufficiently large and cramping the flue area underneath the floor, thus compelling the forcing of fire which not only prevents the obtaining of satisfactory drying results, but burns out fire-box,

grate bars and flues. An excellent fuel for obtaining a quick heat as well as a very high temperature for small box type ovens, such as Millet, Wadsworth, etc., is petroleum coke, the cupola coke being too slow of combustion for the small fire-box.

In building ovens, the proper foundation for building car tracks in ovens is well worth consideration, as Mr. Fuller had had cases when it was found necessary, at considerable trouble and expense, to tear up floors and put in foundations under rails in ovens which were otherwise perfect.

AUGUSTUS N. KELLEY[1] wrote that mixing core sand with both oil and glutrin makes a very good core, if it is to be used immediately, but when allowed to stand in the mold an extra length of time, it seems to accumulate moisture. Several years ago he had difficulty in making some small cores that had to stand a heavy body of iron, but was finally able to accomplish this by using ground asbestos, sand and oil. He also had difficulty in getting new men to ram the sand properly and lifting the box off the core without breaking it. He overcame this by inventing a machine that jar rams the sand, rolls over the core box and plate, and lifts the core box, leaving a perfect core on the plate ready for the oven.

ALEX. E. OUTERBRIDGE, JR.,[2] in a written discussion, said that he hesitated to accept the author's estimate of "a few thousandths of an inch variation." The expansion of the materials composing the core when subjected suddenly to the intense heat of molten metals and the contraction of the metals in cooling from the liquid to the solid state, together with the coarse-grained character of the sands used in many cores, would seem to limit the possible degree of accuracy of cored holes in castings. The use of coremaking machines is now general, if not universal. Large cores are made by machine with unskilled labor, in one-fourth of the time required to make the same cores by hand, and, moreover, ordinary heap sand is used without the addition of loam, flour, or any other binder. For core testing, in place of the crude device shown in Fig. 8 for breaking 1-in. sq. cores with supports 12 in. apart, the ordinary transverse testing machine used for breaking cast-iron test bars has been employed for a number of years by Mr. Outerbridge, with test cores 1 in. by 1 in. by 15 in. In concluding, Mr. Outerbridge called attention to the method described in

[1] Foundry Supt., American Blower Co., Detroit, Mich.

[2] Wm. Sellers & Co., Inc., Phila., Pa.

Volume 11 of the Proceedings of the American Society for Testing Materials, for instantaneous detection of adulterations of linseed oil and other oils used in making oil sand cores.

H. A. BECKER[1] wrote that the core room might be made to serve the foundry in such a way as to minimize molding expense, and he would like to make a few suggestions along this line. Direct labor, that is that of molders or coremakers, is the greatest factor in the

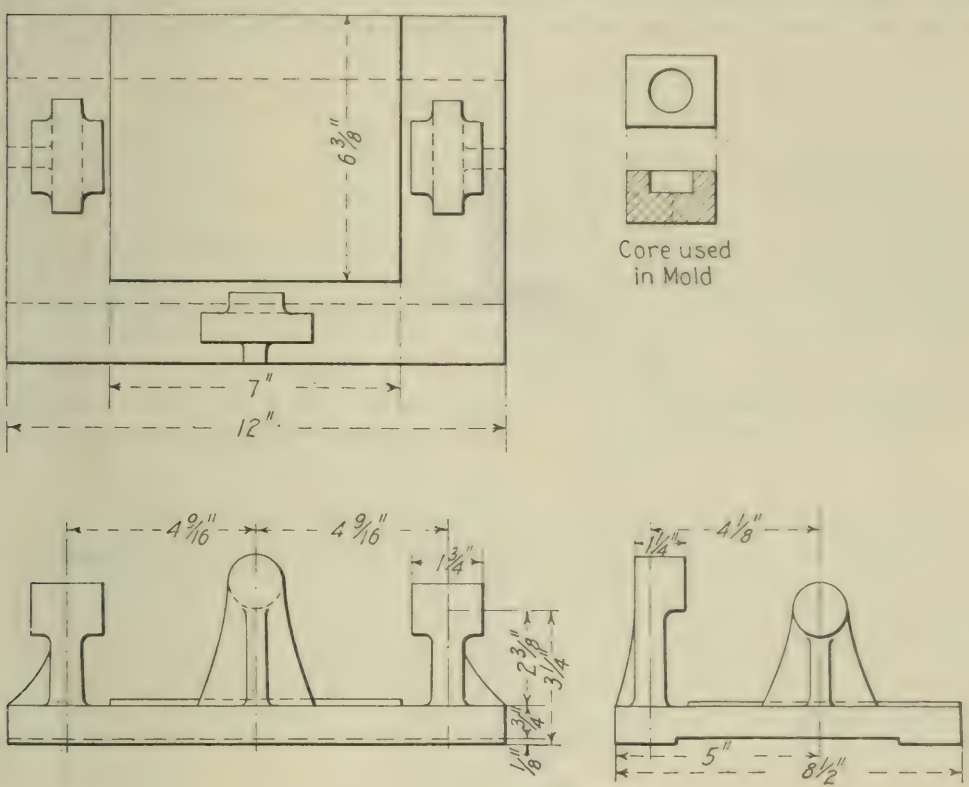


FIG. 45 MACHINE SHOP JIG

total cost of a casting; because in cost-finding methods the running expenses or overhead is usually charged against the direct labor; therefore, to reduce the cost of a casting the time necessary to make it rather than the cost of materials used in it must be reduced.

He had found a very simple method of saving time to be the use of cores to form certain parts of the work in ordinary green sand molds. This class of work could be divided into two parts.

The first is the somewhat intricate casting with drawbacks in the

[1] East Moline, Ill.

drag, in which only one casting is required, as for instance, the machine shop jig shown in Fig. 45. In this case, the bosses forming the ends of the bearings would have to be made loose on the pattern and drawn into the spaces left by the ribs after the pattern was drawn, if the mold were made all in green sand. In the design of some jigs of this kind, it is impossible to draw the bosses out through the ribs because the ribs are not thick enough, and at best it is a tedious and delicate job. By using cores, as shown in the upper right-hand corner of the drawing, a great deal of time can be saved by the molder. Instead of using the boss *A* on the pattern, it is set in an ordinary core box or

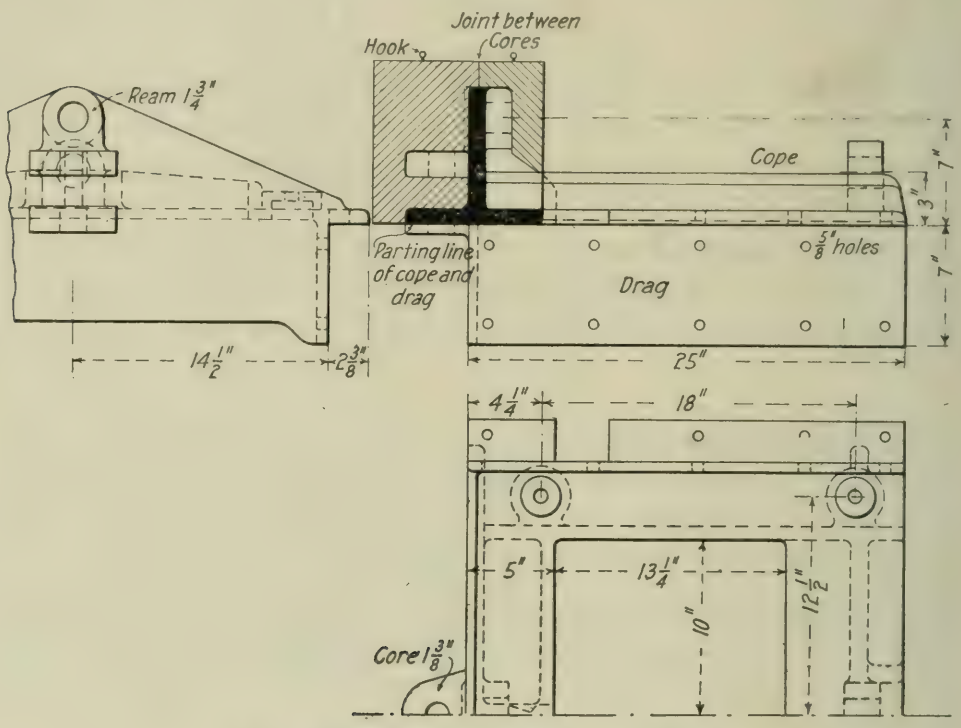


FIG. 46 SAMPLE OF STANDARD WORK

core frame with a dowel pin to center it and the cores made in this box or frame are pinned to the pattern with a loose pin in place of the original boss. Pieces of any shape can be formed in this manner as accurately as they could with the complete pattern. As the sand is rammed about the pattern the loose pins are withdrawn and when the pattern is finally drawn, the cores remain in the sand and form the undercut or drawback portions of the mold.

The second class of work is shown in Fig. 46, which represents a sample of standard work of which there are many castings to make, but the piece is of such an awkward design from the molder's point of

view that it would be difficult to make up in green sand. The section lines show two cores which can be made and set against the pattern and the heavily shaded part of the pattern represents the bearings which were made to support the cores. The core is made in two pieces, as shown, each being a simple block core formed in a core frame or core box. These two block cores can be laid on the pattern so that they fit accurately, without any chance of their displacement, and then tied into the cope with wires attached to the hooks. After the core is rammed up and the pattern drawn, the result is a mold finished in one operation from a pattern that would ordinarily introduce many difficulties in the work.

If the patternmaker studies the possibilities of the use of cores for forming difficult parts of the casting, it will be no more difficult or costly to make the patterns so as to produce the castings by this method than to make them in the ordinary way. A core rammed up in a mold shows practically no joint where it meets the green sand, whereas a core set into place after the mold is made rarely fits the print exactly and is bound to show unevenness along the joints.

Many examples of this class of work might be shown, but these illustrate the principal advantages and will suggest a method of working out such a problem.

Thomas D. West emphasized the necessity of grinding the sand properly, and told of a case where the grinder was out of commission and an attempt was made to prepare the mixture by shoveling, tramping it with the feet, and pounding with rammers. A large amount of labor was expended in this hand treatment, but very little success achieved in securing the plastic body obtained by the grinding process.

Richard Moldenke called attention, in a written contribution, to the lamentable lack of initiative in all departments of the foundry, particularly in the core room. He had frequently been called into foundries where difficulties in practice existed, only to find that the foremen were insufficiently informed concerning the principles underlying their work. He urged the placing of Mr. Lane's paper in the hands of core-room foremen, with instructions to repeat at least the leading tests made by Mr. Lane. An establishment in which the testing out of new lines of operation is encouraged will always be at the front of the procession, and Mr. Lane has shown the way for the core room.

THE AUTHOR said with regard to the query of Mr. Outerbridge, that there are several industries in which cast fits are regularly made. In the lock industry the lock cases themselves are made to fit with very little grinding, and in some cases with none at all. As to the necessity of grinding the material, since the paper was written, he had run tests in a number of foundries on both green sand and dry sand facing mixtures. In these mixtures the bond always consists of clay as a base, and there is usually present either flour or dextrin, or glutrin or molasses water. The grinding of clay and glutrin together develops the maximum colloidal condition, and hence bonding efficiency. The grinding of the clay with the flour or dextrin tends to develop a perfect paste, and this is distributed among the sand grains in the most efficient manner by grinding. The surface of each grain is scoured and the bonding material brought into absolute contact with the grain. The grinding also forces the sharp points of the grains to pass one over each other, so that the grains themselves may be said to serve as small shovels or paddles between which the bonding material is being worked and kneaded.

No. 1333

TESTS OF A SAND-BLASTING MACHINE

By Wm. T. Magruder, Columbus, Ohio

Member of the Society

This paper gives the records, results and conclusions of a series of quantitative tests of a sand-blasting machine, under the actual conditions of commercial practice, which were made by David H. Ebinger and Robert A. Frevert, of Columbus, under the direction of the author, as their thesis for the degree of mechanical engineer from The Ohio State University in 1910. The variables in the problem were: (*a*) the material to be sand-blasted; (*b*) the character of its surface; (*c*) the air pressure best to use with different materials and conditions; (*d*) the size of nozzle for different classes of work; (*e*) the angle to the surface of the work at which the nozzle should be held; (*f*) the distance from the work at which the nozzle should be held; (*g*) size, sharpness, kind, character, uniformity of size, and cleanliness of the sand, and the number of times it has been previously used for sand-blasting; (*h*) the relative dryness of the sand and of the compressed air of the blast; (*i*) the proportion of sand to air; (*j*) differences in commercial machines.

2 As it was evident that a complete solution of the problem was impossible in the time available, certain of the variables were made constant as follows:

a The material was pieces of cast iron which had been broken as test bars for transverse tests. They were 2 in. by 4 in. in cross-section and from 15 to 25 in. long. They had been cast diagonally on edge in one mold from one ladle of machinery iron.

b Their rough surfaces were as uniform as were the bars themselves.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

c A new nozzle, $\frac{5}{16}$ in. in diameter, was used for each test. To remove any unevenness of the interior surface of the nozzles, and to bring them all exactly to the same condition, sand was blown through each for 2 minutes previous to starting the tests. No tests have yet been made with other sizes of nozzles.

d The sand used was a No. 3-J Cape May grit, new, hard, sharp, clean, free from clay, and was such as is commonly used by the trade. It was obtained from a large storage bin in the works. It was thoroughly dried in a Pang-

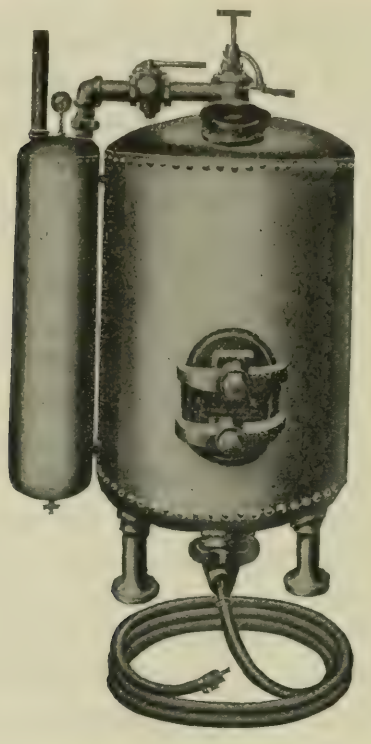


FIG. 1 SAND-BLASTING MACHINE

born No. 1, Type M, sand dryer, shortly before being used. Great care was taken to avoid overheating it and so destroying its strength and cohesion. The sand was then passed through a No. 8 mesh screen. After use in the machine, it was weighed, sifted, reweighed, and discarded. New sand was used for each test.

e The sand was dried as above described. The compressed air was passed through a separator to relieve it of moisture from the atmosphere and oil from the compressor.

f No attempt was made to regulate and adjust the propor-

tion of air to sand. All tests were run with the regulating valve wide open.

g No comparisons were made of the operations or results from different machines on the market.

3 Upon the three remaining variables, the air pressure, the angle between the surface of the work and the nozzle, and its distance from the work, the quantitative experiments were made.

EQUIPMENT

4 The regular equipment of the D. A. Ebinger Sanitary Manufacturing Company at Columbus, Ohio, was used for these tests. It

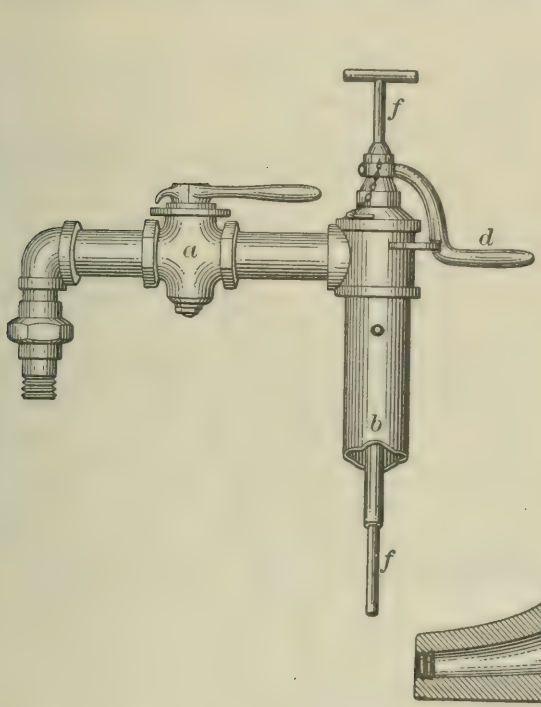


FIG. 2 REGULATING AND CONTROLLING VALVES

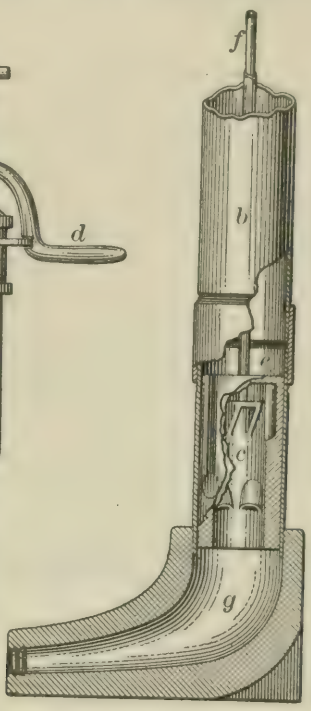


FIG. 3 MIXING CHAMBER FOR AIR AND SAND

had been recently installed for use in sand-blasting cast-iron ware before enamelling. It consisted of one No. 8, Type C, sand-blasting machine, made by the Thomas W. Pangborn Company, Jersey City, N. J., having a sand capacity of 4000 lb. per charge. It was provided with a moisture separator, pneumatic sand separator, dust catcher, and exhaust fan. The air was compressed by a Fairbanks, Morse & Company air compressor, having a capacity of 138 cu. ft. of free air per minute, and driven by an opposed gas engine of 25 h.p capacity.

5 Fig. 1 shows the machine and Fig. 2 the regulating and controlling valves. Fig. 3 shows the mixing chamber for the air and sand. Fig. 4 shows a plan of the rooms and apparatus. Fig. 5 shows the sand-blasting box and sand-catcher. Fig. 6 shows a photograph of the machine, blasting room and apparatus. Table 1 gives the record of the tests and the results. Figs. 7 to 9 graphically show the results obtained.

OPERATION OF THE MACHINE

6 The compressed air, coming from the compressor, enters the moisture and oil separator at the left of the machine, passes through

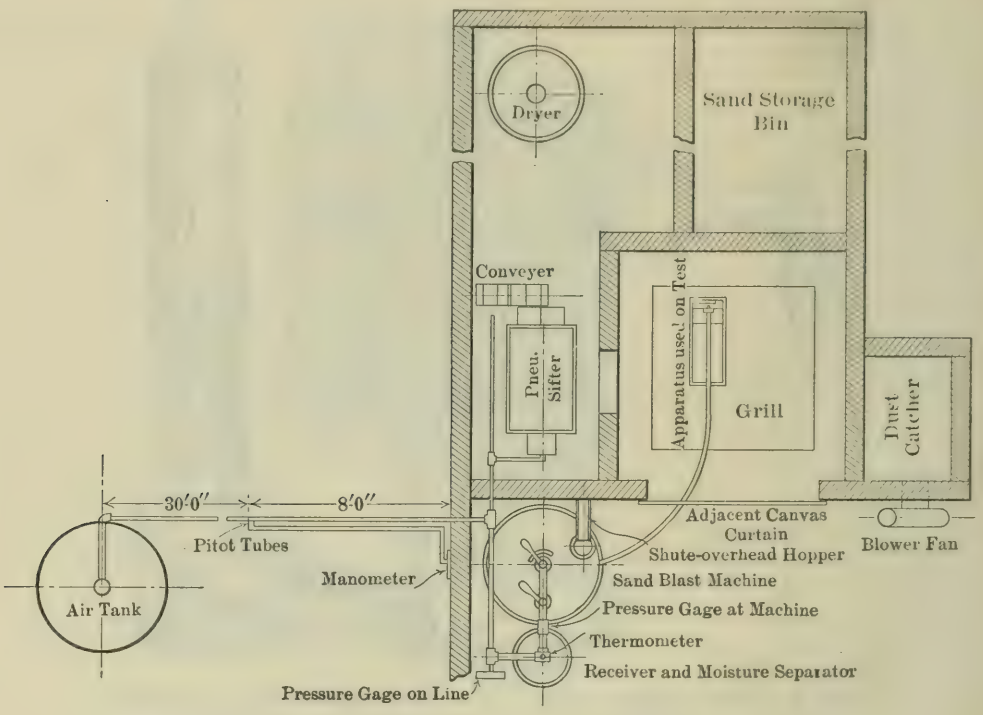


FIG. 4 LOCATION AND ARRANGEMENT OF APPARATUS

the air regulator *a* (Fig. 2), which has an indicator-handle working over a graduated disc, enters and passes down through the cylinder *b*, inside the machine, then through the air ports in the piston *c* (Fig. 3), and engages the entering sand. The ports in the top of the cylinder, one of which is shown, deliver the air to the sand chamber, so as to maintain equal pressure above the sand and to assist in insuring uniform flow. The sand-controller handle *d* moves on a quadrant, having limit stops for its off and on full positions. This handle permits the piston *c*

to be rotated in its casing *e*, thereby opening, regulating and closing the sand ports by a single control. The stirrer and handle *f* connect with its fork which operates inside of the piston and mixing chamber for dislodging caked sand that may form under certain conditions. Ports, elliptical in shape, are located in opposite sides of both the piston *c* and the casing *e*, and are inclined downwards, so starting the flow of the sand by gravity. The rotation of the piston by the controller handle regulates the opening of the ports and the flow of sand. The sand on leaving the ports is met by the air coming through the ports in the piston. They cross each other from opposite sides into the mixing chamber *g*. By this cross flow, a swirl-

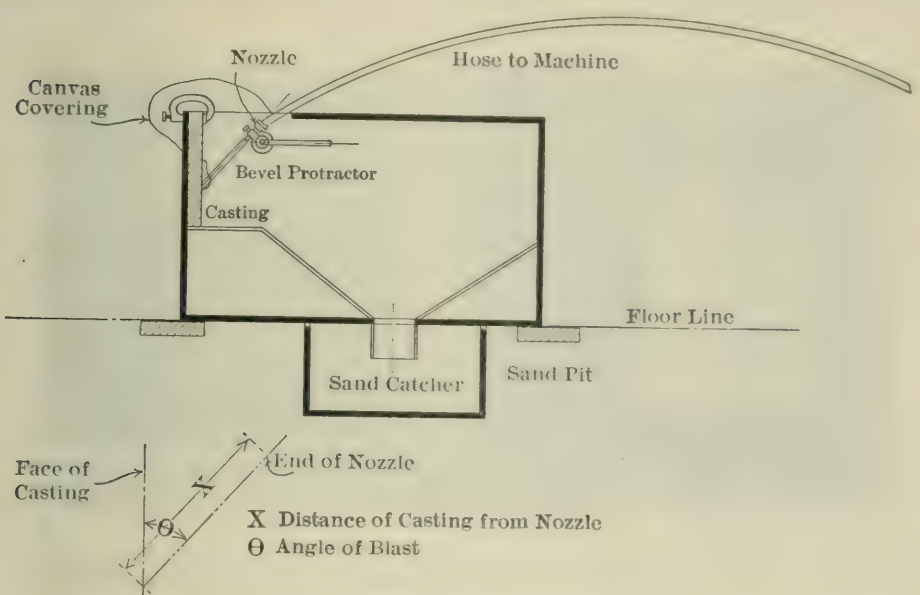


FIG. 5 SAND-BLASTING BOX AND SAND CATCHER

ing motion is produced and a thorough mixture of the sand and air is obtained in the rear part of the mixing chamber. The mixture is carried forward by the air pressure from the rear part, $2\frac{1}{2}$ in. in diameter, to the hose connection where it is $\frac{3}{4}$ in. in diameter, thence through the hose to the nozzle, whence it is projected upon the casting to be sand-blasted.

MEASUREMENT OF THE AIR

7 The air used was measured by a pitotmeter placed horizontally in a run of straight $1\frac{1}{2}$-in. standard welded pipe in the main air line, and located at a distance from any fitting. The opening

of the dynamic tube, $\frac{1}{8}$ in. in diameter, faced the current of air from the receiver tank. The end of the static tube was trimmed off flush with the interior of the pipe. The tubes were connected by $\frac{1}{8}$-in. pipes to the ends of a water manometer. The difference in the levels of the liquid in the two legs of the manometer indicated the difference between the dynamic and static pressures, and is a measure of the velocity head in inches of water. From the formula, $v^2 = 2gh$, the velocity of the air in the pipe was calculated. Deducting the area of the dynamic tip from the area of the opening in the pipe, gave the net area of the air pipe. The coefficient of discharge was taken as 0.91. To remove the moisture that collected in the tubes, valves were placed in each of the tubes above the manometer, and pet cocks

FIG. 6 VIEW OF MACHINE AND APPARATUS IN BLASTING ROOM

placed in the tubes directly above the valves. This arrangement permitted the blowing out of each tube before taking a reading of the pitotmeter. The temperature of the air in the line was taken just before it entered the stop valve at the machine. Calibrated pressure gages in the line and at the machine gave the desired pressures.

MEASUREMENT OF THE SAND

8 To collect the sand used during each test and also to control the blast, a closed box with a hopper bottom and a shelf at one end to support the test bar was prepared, as is shown in Fig. 5. The sand discharged during a test was collected in the hopper, removed, and weighed as total sand used. It was then sifted, and again weighed, thus giving by difference the amount of sand rendered useless by the

test, and the amount of sand that might be used again in commercial practice. It was not again used in these tests.

MEASUREMENT OF THE IRON REMOVED

9 The test bar was held in a vertical position in the sand-blasting box. Before and after blasting, it was carefully weighed on a plat-

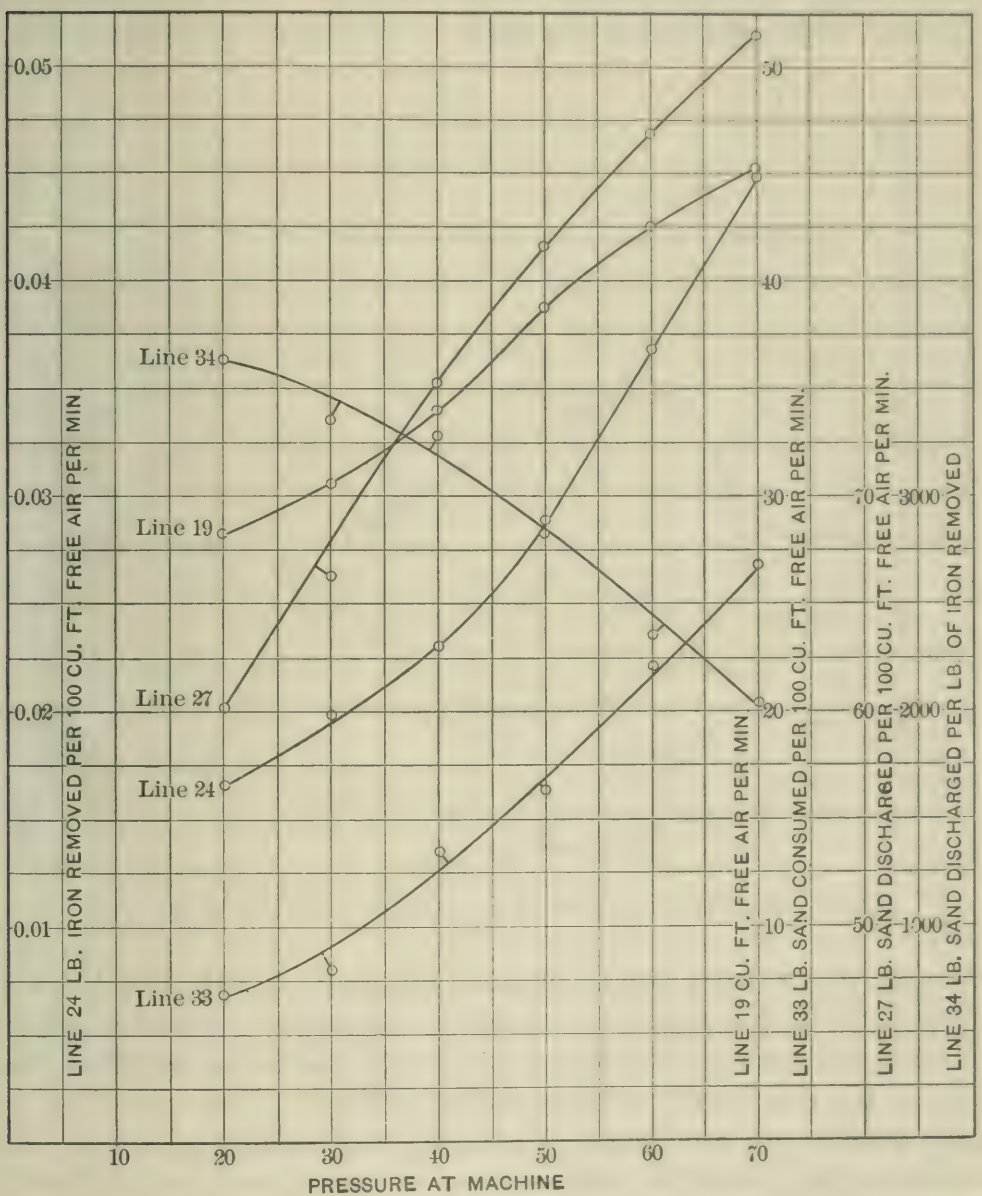


FIG. 7 RESULTS OF TESTS IN RELATION TO AIR PRESSURE AT MACHINE

form scale of 250 lb. capacity by quarter ounces. The scale was carefully balanced before each weighing.

METHOD OF MAKING THE TESTS

10 In each test the new nozzle was first brought to a uniform condition by discharging a blast of sand and air through it for 2 minutes under a constant blast pressure. The test bar was weighed and placed in position in the closed hopper box. The nozzle was then set by scale for its distance from the test bar and by bevel protractor for its angle with the surface of the test bar. The air regulator *a* was kept wide open during the tests. The desired air pressure in the

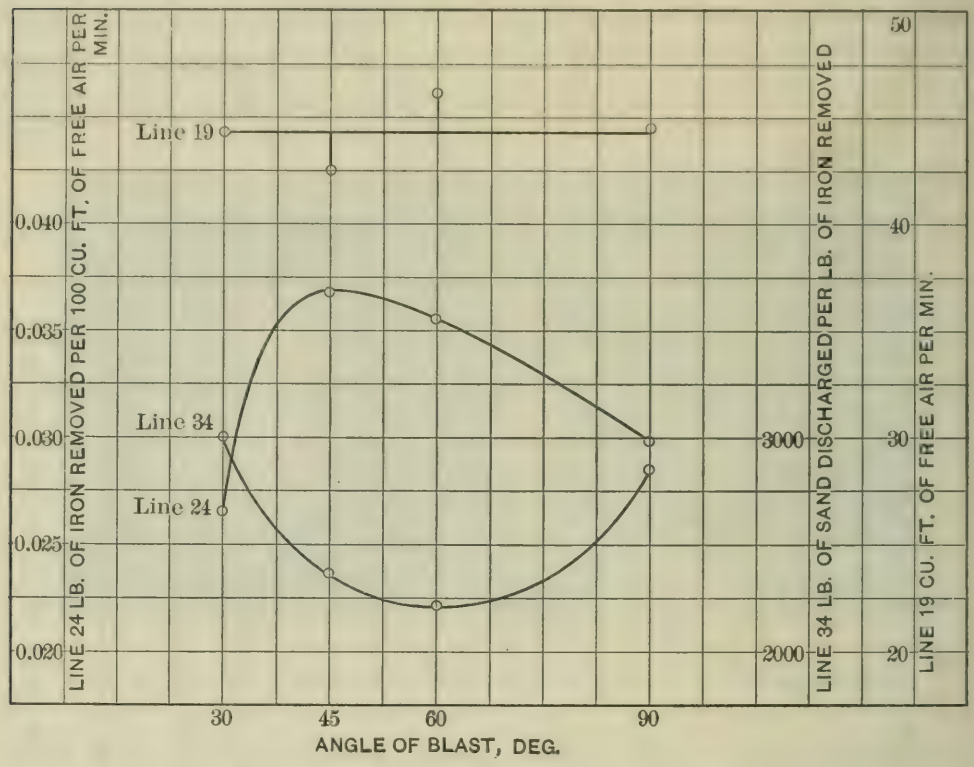


FIG. 8 RESULTS OF TESTS IN RELATION TO ANGLE OF BLAST

machine was secured and maintained constant by adjusting the stop valve in the supply pipe to the moisture separator. After starting the air blast and carefully adjusting the valve to maintain a constant pressure at the machine, the sand-controller handle *d* was moved to its open position at the instant of starting, thus allowing sand to flow into the mixing chamber and be forced by the blast of air out through the nozzle and against the test piece. Readings of the temperature and pressures of the air on the main line and at the machine were taken at regular intervals of about 2 minutes. In closing down the machine at the end of a test, the sand valve was first closed and

then the air valve; the test bar was weighed, and the difference of the two weights gave the amount of iron removed; the sand discharged was weighed, sifted and again weighed.

VARIABLES OF THE TESTS

11 Starting at 20 lb. corrected pressure at the machine on the first test, and increasing by 10 lb. up to and including 70 lb. pres-

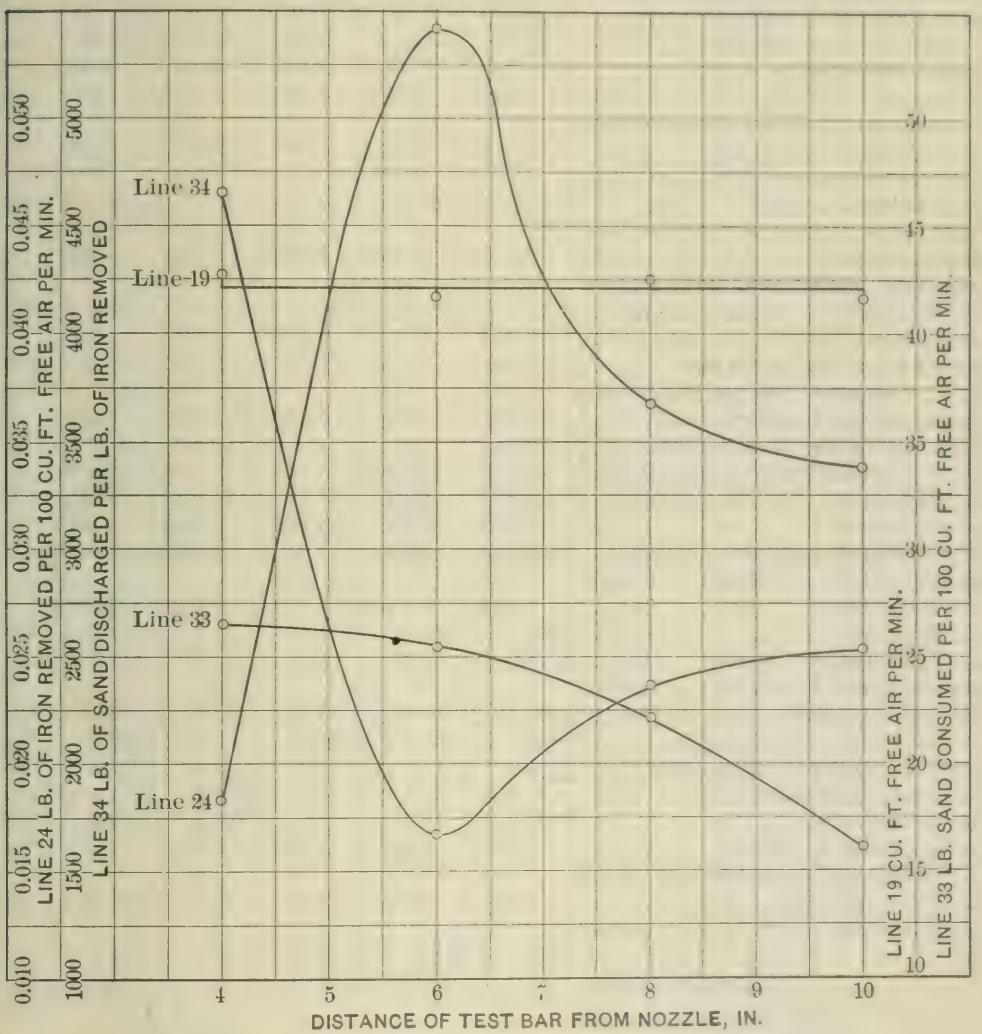

Line 34
Line 19
Line 33
Line 24
LINE 24 LB. OF IRON REMOVED PER 100 CU. FT. FREE AIR PER MIN.
LINE 34 LB. OF SAND DISCHARGED PER LB. OF IRON REMOVED
LINE 19 CU. FT. FREE AIR PER MIN.
LINE 33 LB. SAND CONSUMED PER 100 CU. FT. FREE AIR PER MIN.
DISTANCE OF TEST BAR FROM NOZZLE, IN.

FIG. 9 RESULTS OF TESTS IN RELATION TO DISTANCE OF NOZZLE FROM WORK

sure, the variation in the effectiveness of the blast due to increase of pressure was obtained, with the nozzle set at 45 deg. and at 8 in. from the test bar.

12 In the second series of tests, a constant pressure of 60 lb. was

TABLE 1 TESTS OF SAND-BLASTING MACHINE

		Pressure Variable					
1	Number of test	1	2	3	4	5	6
2	Diameter of nozzle, in.	5/16	5/16	5/16	5/16	5/16	5/16
3	Angle of blast, deg.	45	45	45	45	45	45
4	Distance of nozzle from casting, in.	8	8	8	8	8	8
5	Length of run, min.	15	13	20	20	10	10
6	Air pressure at sand-blasting machine, lb.	20	30	40	50	60	70
7	Air pressure in line, lb., corrected	78.4	75.0	76.1	75.0	75.0	76.5
8	Barometric pressure, in. mercury	29.08	29.13	29.35	29.35	29.30	28.80
9	Temperature of air, deg. fahr.	77.6	85.1	82.00	83.20	87.50	80.40
10	Pitotmeter reading, in. water	0.056	0.070	0.085	0.112	0.135	0.150
11	Velocity head, ft. water	0.0047	0.0058	0.0071	0.0093	0.0113	0.0125
12	Volume as free air of 1 cu. ft. of compressed air under actual conditions	5.796	5 508	5.607	5.527	5.483	5.618
13	Weight of 1 cu. ft. of water at temperature of air in line	62.25	62.18	62.21	62.20	62.16	62.23
14	Weight of 1 cu. ft. air at actual pressure and temperature	0.4679	0.4446	0.4526	0.4462	0.4427	0.4560
15	Head of compressed air equivalent to 1 ft. of head of water	133.0	139.9	137.4	139.4	140.4	136.5
16	Velocity head, ft. air	0.6253	0.8111	0.9758	1.296	1.567	1.706
17	Velocity in air line, ft. per sec.	6.342	7.223	7.922	9.132	10.10	10.47
18	Cu. ft. compressed air per min. at line pressure and temperature	4.866	5.543	6.079	7.007	7.752	8.038
19	Equivalent free air, cu. ft. per min.	28.21	30.53	34 09	38.73	42.51	45.40
20	Weight of test bar before blasting, lb.	44.5469	41.3906	36.4531	41.6094	41.3125	28.7031
21	Weight of test bar after blasting, lb.	44.4765	41.3125	36.2969	41.3906	41.1563	28.5000
22	Iron removed, lb.	0.0704	0.0781	0.1562	0.2188	0.1562	0.2031
23	Iron removed, lb. per min.	0.00469	0.00601	0.00781	0.01094	0.01562	0.02031
24	Iron removed, lb. per 100 cu. ft. of free air flowing per min.	0.01663	0.01969	0.02291	0.02825	0.03675	0.04473
25	Sand discharged, lb.	255.0	262.0	512.0	631.0	369.0	415.0
26	Sand discharged, lb. per min.	17.0	20.2	25.6	31.5	36.9	41.5
27	Sand discharged, lb. per 100 cu. ft. of free air flowing per min.	60.27	66.19	75.10	81.34	86.81	91.41
28	Usable sand remaining, lb.	226.0	230.0	420.0	504.0	275.0	293.0
29	Usable sand remaining, lb. per min.	15.1	17.7	21.0	25.2	27.5	29.3
30	Usable sand remaining, per cent.	88.6	87.8	82.0	80.0	74.5	70.6
31	Sand consumed, lb.	29.0	32.0	92.0	127.0	94.0	122.0
32	Sand consumed, lb. per min.	1.93	2.46	4.60	6.35	9.40	12.20
33	Sand consumed, lb. per 100 cu. ft. of free air flowing per min.	6.843	8.059	13.50	16.40	22.11	26.87
34	Sand discharged per lb. of iron removed, lb.	3625.0	3361.0	3278.0	2879.0	2362.0	2043.0

Lines 6 and 7 give corrected gage readings.

Line 12 is obtained from the characteristic equation for perfect gases $V_1 = \frac{492}{14.7} \times \frac{P}{T}$, by taking the absolute pressures and temperatures corresponding to the readings given on Lines 7 and 9.

Line 13 is taken from p. 688 of Kent's Pocket Book for the temperatures given on Line 9.

Line 14 = Line 12 × weight of 1 cu. ft. free air.

$$\text{Line 15} = \frac{\text{Line 13}}{\text{Line 14}}$$

Line 16 = Line 15 × Line 11

$$\text{Line 17} = 8.02 \sqrt{\text{Line 16}}$$

Line 18 = Line 17 × net area of pipe in sq. ft. × coefficient of discharge × 60

Line 19 = Line 18 × Line 12

$$\text{Line 23} = \frac{\text{Line 22}}{\text{Line 5}}$$

$$\text{Line 24} = \frac{100\ (\text{Line 23})}{\text{Line 19}}$$

COLLECTED DATA AND CALCULATED RESULTS

	Angle Variable			Distance Variable				
7	8	9	10	11	12	13	14	7A
$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$
30	45	60	90	45	45	45	45	30
8	8	8	8	4	6	8	10	8
8	10	10	10	10	9	10	10	10
60	60	60	60	60	60	60	60	60
78.1	75.0	77.0	79.0	78.4	77.7	75.0	77.0	77.8
29.13	29.30	29.20	29.20	20.08	29.30	29.30	29.08	29.30
81.60	87.50	78.10	80.15	78.97	89.66	87.50	77.94	79.44
0.140	0.135	0.154	0.140	0.130	0.127	0.135	0.125	0.208
0.0117	0.0113	0.0128	0.0117	0.0108	0.0106	0.0113	0.0104	0.0173
5.735	5.483	5.704	5.806	5.781	5.626	5.483	5.705	5.739
62.21	62.16	62.25	62.23	62.24	62.13	62.16	62.25	62.24
0.4630	0.4427	0.4604	0.4687	0.4667	0.4542	0.4427	0.4606	0.4633
134.4	140.4	135.2	132.8	133.04	136.8	140.4	135.2	134.3
1.572	1.587	1.730	1.553	1.440	1.450	1.587	1.406	2.324
10.06	10.10	10.55	9.996	9.625	9.657	10.10	9.508	12.23
7.717	7.752	8.096	7.670	7.386	7.411	7.752	7.296	9.382
44.25	42.51	46.18	44.53	42.70	41.69	42.51	41.63	53.84
36.5313	41.3125	51.6172	30.8437	44.2656	41.1250	41.3125	44.4062	41.4219
36.4375	41.1563	51.4531	30.7109	44.1875	40.9219	41.1563	44.2656	41.3125
0.0938	0.1562	0.1641	0.1328	0.0781	0.2031	0.1562	0.1406	0.1094
0.0117	0.01562	0.01641	0.01328	0.00781	0.02257	0.01562	0.01406	0.01094
0.02644	0.03675	0.03554	0.02982	0.01829	0.05413	0.03675	0.03378	0.0203
281.0	369.0	364.0	378.0	361.0	341.0	369.0	356.0	243.0
35.1	36.9	36.4	37.8	36.1	37.9	36.9	35.6	24.3
79.32	86.81	78.83	84.88	84.55	90.90	86.81	85.52	45.13
218.0	275.0	268.0	279.0	248.0	246.0	275.0	289.0	180.0
27.3	27.5	26.8	27.9	24.8	27.3	27.5	28.9	18.0
77.58	74.5	73.6	73.8	68.7	72.1	74.5	81.2	74.1
63.0	94.0	96.0	99.0	113.0	95.0	94.0	67.0	63.0
7.875	9.4	9.6	9.9	11.3	10.6	9.4	6.7	6.30
17.80	22.11	20.79	22.23	26.46	25.42	22.11	16.09	11.70
3000.0	2362.0	2218.0	2846.0	4642.0	1679.0	2362.0	2532.0	2221.0

$$\text{Line 26} = \frac{\text{Line 25}}{\text{Line 5}}$$

$$\text{Line 27} = \frac{100\ (\text{Line 26})}{\text{Line 19}}$$

$$\text{Line 29} = \frac{\text{Line 28}}{\text{Line 5}}$$

$$\text{Line 30} = \frac{\text{Line 29}}{\text{Line 26}}$$

$$\text{Line 32} = \frac{\text{Line 31}}{\text{Line 5}}$$

$$\text{Line 33} = \frac{100\ (\text{Line 32})}{\text{Line 19}}$$

$$\text{Line 34} = \frac{\text{Line 27}}{\text{Line 24}}$$

maintained at the machine, and the nozzle set at 8 in. from the test bar. The angle was varied from 30 to 90 deg. The effectiveness of these angles of blast was in this way obtained.

13 In the third series of tests, the distance of the nozzle from the test piece was varied, the angle being set at 45 deg. and the air pressure maintained constant at 60 lb.

OBSERVATIONS

14 It was noted that with no sand flowing, the quantity of air flowing approximated the theoretical discharge for the nozzle. On turning on the sand, the quantity of air flowing immediately decreased; and when the sand was "on full," the amount of air discharged was only 40 to 50 per cent of the original quantity.

15 With constant air pressure and with the sand-controller lever in its "on full" position, on account of the variation in the openings of the sand and air ports of the machine, about 30 tests were required to be made in order to get even fairly concordant results for the pounds of sand discharged per 100 cu. ft. of free air flowing per minute, as shown on Line 27, Table 1. For constant pressure, variations in the pitotmeter readings were due to the variations in the sand flowing. As the quantity of sand decreased, the velocity and quantity of air increased, but the quantity of sand discharged per pound of iron removed also decreased. This can be seen by comparing the readings on Line 34 for tests 7 and 7*A*. It would seem that further experiments might prove that at other angles than 30 deg., and at other distances than 8 in., and at other pressures than 60 lb., less sand would be used per pound of iron removed if the sand-regulating valve was not open so much. This would make the energy of the individual grain of sand greater and its blasting effect larger. Furthermore, there would be less piling up of the sand on the surface being sand-blasted.

RESULTS

16 From these quantitative experiments on a cast-iron test bar for these three variables, within the limits used, the results as given in Table 1 and graphically shown in Figs. 7, 8 and 9, are as follows:

a For a constant distance of 8 in., and at a constant angle of 45 deg., between the nozzle and the test bar, the equivalent amount of free air delivered per minute, the iron removed, the sand discharged, and the sand used up per

100 cu. ft. of free air flowing per minute, vary directly with the pressure; the per cent of usable sand remaining, and the amount of sand discharged per lb. of iron removed vary inversely with the air pressure in the machine. (See Lines 19, 24, 27, 33, 30, and 34, Fig. 7.)

b With a constant pressure of 60 lb. in the machine, and a fixed distance of 8 in. from the nozzle to the test bar, the largest amount of metal was removed, and the least amount of sand was required to do it, when the angle between the nozzle and the surface of the work was from 40 to 60 deg. (See Lines 24 and 34, Fig. 8.)

c With a constant pressure of 60 lb. in the machine and a constant angle of 45 deg. between the nozzle and the surface of the test bar, the largest amount of metal was removed, and the least amount of sand was required to do it, when the distance from the nozzle to the work was about 6 in. (See Lines 24 and 34, Fig. 9.)

d With a constant pressure of 60 lb. and a fixed distance of 8 in. from the nozzle to the test bar, the amount of sand used up varies as the directness of the blast. (See Line 33, Table 1.)

e With a constant pressure of 60 lb., and a constant angle of 45 deg. between the nozzle and the test bar, the amount of sand used up varies inversely with the distance of the nozzle from the test bar. (See Line 33, Fig. 9.)

f For a constant angle of 45 deg., and a fixed distance of 8 in. between the nozzle and the test bar, twice as much metal was removed at 56 lb. pressure as at 20 lb.; at 64 lb. as at 30 lb.; and at 72 lb. pressure as at 40 lb. (See Line 24, Fig. 7.)

g With a constant pressure of 60 lb., and a constant distance of 8 in. between the nozzle and the test bar, over 20 per cent more metal was removed with the nozzle held at 45 deg. than at 90 deg. to the test bar.

h With a constant pressure of 60 lb., and a constant angle of 45 deg. between the nozzle and the test bar, 60 per cent more metal was removed when the nozzle was held at 6 in. than at 10 in. from the test bar.

CONCLUSIONS

17 Unless it can be shown that the extra cost of compressing one hour's supply of air, or 2760 cu. ft. of free air per hour (60 x 46 cu. ft. per minute for 72 lb. pressure, line 19 of Test 6) to a pressure of 72 lb. per sq. in. exceeds the cost of compressing two hours' supply of air, or 4090 cu. ft. of free air (2 x 60 x 34.09 cu. ft. per minute for 40 lb. pressure, line 19, Test 3) to 40 lb. by the cost of an hour of labor (25 cents), the higher pressures, delivered at an angle of about 45 deg., and at a distance of about 6 in. from the work, are to be preferred for the sand-blasting of cast iron.

18 Assuming that the actual horsepowers required to compress 1 cu. ft. of free air per minute to be 1.1 times that theoretically required, or 0.113 to compress to 40 lb. and 0.165 h.p. to compress to 72 lb. gage pressure,[1] the 34.09 cu. ft. per minute at 40 lb. will require 7.7 h.p-hr., and the 46 cu. ft. per minute at 72 lb. pressure will require 7.59 h.p-hr. It will thus be seen that the total power required to compress the necessary air to remove 1 lb. of iron is practically the same, irrespective of the air pressure.

19 If this should be true for other relative pressures and conditions, the total cost of sand-blasting equals the constant cost of compression per pound of iron removed, plus the sum of the time-costs of labor, interest and overhead charges; and therefore, the higher the pressure used, the less will be the time, labor, and total cost of removing a pound of material.

DISCUSSION

F. C. Brooksbank[2] observed that no test was made of low pressures. The pressures seem to run between 20 and 70 lb. There is a great amount of work that can be done at 5 and 10 lb. and 25 lb. is sufficient to do any kind of work. The most valuable part of the sand blast is to have a large volume of air at the low pressure and carry a larger amount of sand. He did not think it necessary to use 50, 60 or 70 lb. when 20 or 25 lb. would do the work, but if the machine was not efficient and must have 20 to 40 lb., that was another matter. There are, however, machines which will work at 5 lb. and there is a great amount of work to be done at that pressure in a better way than at high pressures. The difference between the cost of compressing air at 15 to 20 lb., or 60 to 70 lb. is a totally different thing.

[1] Kent's Pocket Book, p. 606.
[2] Tilghman–Brooksbank Sandblast Co., Phila., Pa.

W. S. GIELE said that two years ago he had tried to get some definite, accurate information from the manufacturers of sand-blasting apparatus as to the conditions on which the efficiency of such an apparatus depends. He found that many foundrymen had the idea that sand-blasting was a scouring operation. In his opinion it was a matter of hammer blow rather than scouring and the desired action is to impact such kinetic energy of the impinging particles of sand that they, striking the casting, will clean off the scale.

In the selection of apparatus his observation had been that probably the most efficient apparatus would be that which would impart the greatest kinetic energy to the sand stream with any given sand, that is, if the particles of sand are all of a given weight, the energy will be proportioned to the square of the velocity of the particles, and therefore the effectiveness of the apparatus considered separately varies directly as the velocity which it imparts to the sand.

He supposed that that sand which had the largest individual grains the apparatus would handle, the largest proportion of grains uniformly of that size and the highest specific gravity, would be the most efficient, the efficiency of operation varying directly, of course, with the energy of the individual grains. The least friability of grains would also be an important point to consider in selecting the sand, as the least friable grains would waste less of their kinetic energy in the breaking up of the sand itself.

In the selection of the distance between the nozzle and the surface of the casting being cleaned and of the angle of impact, there is greater room for variation than in any of the other conditions. This depends, of course, on the operator and on the shape and position of the casting upon which he is working.

The efficiency then depends on the apparatus, in so far as the energy imparted to the sand depends upon its velocity; on the sand, in so far as the energy imparted to it by the apparatus depends on the weight of the individual grains; and on the amount of energy dissipated in destruction of the sand itself; the operator being left to determine the distance between the nozzle and the surface of the casting, and also the angle of impact. The effective working force of the operation arises from the change of direction of motion imparted to the sand grains.

The reason why the sand blast is more effective when applied at an oblique angle rather than normal to the surface to be cleaned, lies in the interference between the rebounding grains and the impinging grains of sand.

A good point brought out by the author, often overlooked by many sand-blast operators, is the protection of the casting by the sand which accumulates on its surface. The casting should always be set up or held in such position that the sand falls away as it accumulates, and leaves the surface free for the action of the sand coming later.

A. G. WARREN[1] said that the object of the sand blast is to clean the castings and not to decrease the weight materially. He had always contended that it was useless to use a higher pressure than necessary to remove the sand and scale instantly. A properly designed machine with ample passages for air, such as the right sizes of pipes, etc., gives the whole available pressure at the nozzle and does not make the sand-blasting machine a reducing valve. He regarded the bath-tub industry as the best example of the cleaning of cast iron. It was the first, he believed, to take hold of the sand-blasting machines and make a commercial success of them. The larger number of these manufacturers use a pressure varying from 15 to 25 lb., and with nozzles running from $\frac{5}{8}$ to 1 in. in diameter, which enables them to clean inside and out in less than five minutes' time a tub having some 65 sq. ft. of surface. He had not yet been able to find a so-called high-pressure sand-blasting machine that could equal this record. In the cleaning of steel castings, there is a possible chance of higher pressures, in some lines of work, being used to advantage.

J. M. BETTON[2] had noticed in some applications of sand-blasting, especially when the jet is perpendicular to the work so that the sand seems to boil as if it were a stream of water coming up in a sandy-bottomed stream, and the particles work around in a small circle, finally disperse and are followed by others.

S. C. SMITH suggested that nothing had been said about the standardization of sand, except that at Cape May, and asked the author to make a statement regarding the average size of grains and their shape, also the weight of the grains. He thought this would be a valuable addition to the subject and enable those who do not know much about sand to have a better idea of it.

THE AUTHOR. Before beginning this series of experiments, a search was made of engineering literature to discover what information was

[1] J. W. Paxson Co., Phila., Pa.
[2] 14 Park Place, New York.

available. Such as was found was of a trade character, and descriptive of certain commercial machines. The manufacturers were communicated with for information and advice, and although they kindly replied, few data that were definite and many that the experiments seemed to disprove were received. For example, one manufacturer advised that the nozzle should be held 90 deg. from the casting, as being most efficient.

In reply to the discussion, the power required to compress the necessary air to remove 1 lb. of iron is practically the same, irrespective of the pressure, and the cost of the labor required to remove 1 lb. of iron using 40 lb. pressure is double that required to remove 1 lb. using 72 lb. pressure. As the cost of the labor required to do the same work is $2\frac{3}{4}$ that when using 20 lb. pressure as when using 72 lb., it did not seem desirable to push the comparison any further and measure the metal removed at air pressures under 20 lb. However, it is possible that for the sand-blasting of certain kinds of castings, other elements may enter into the problem which would offset the advantage of the cost of labor of the high-pressure system, and cause the lower pressure to be preferred. The operator's fear of the blast and the damage it could do, caused one sand-blasting equipment to be very inefficient and almost useless; and when the air-valve was opened on him, he promptly resigned.

Replying to Mr. Smith, I would say that the No. 3–J Cape May grit which was used had been passed through a No. 8 mesh screen, after being dried. The grains were well-rounded ovoids. Only a trace passed a No. 30 screen, 3 per cent passed the No. 20 but not the No. 30 screen, 77 per cent passed the No. 10 but not the No. 20 screen, and 20 per cent passed the No. 8 but not the No. 10 screen. In short, 97 per cent was a No. 8–20 sand. Its specific gravity was 2.649. Its fusion point was 3254 deg. fahr. Chemical analysis of a 100-gram sample gave

Loss by ignition, combined water	0.22
organic matter	0.00
Fe_2O_3	0.44
Al_2O_3	0.60
CaO	0.34
MgO	0.31
SiO_2	98.06
Total	99.97

The nozzles were $\frac{5}{16}$ in. in diameter of opening; and, when using this sand, lasted from two to six hours of continuous use, averaging four hours, depending upon their hardness.

No. 1334

DIE CASTINGS

By Amasa Trowbridge, Hartford, Conn.

Member of the Society

The art of die casting, that is, the making of castings in steel molds to finished size and shape, was first introduced in connection with the manufacture of type. The first attempt at type founding was made in France, followed shortly by manufacturers in other countries. The first successful attempt in the United States was about 1735. As soon as it was established that finished type could be cast in steel molds, inventors naturally turned their attention to machines for doing the work. Such machines were attempted early in the 19th century and by the close of that century the perfected linotype and monotype machines, which are fine examples of die-casting machines, had been evolved. Machines for casting individual pieces of type are also still in use and the most recent ones will cast about 125 to 150 pieces per minute. This gives some idea of the speed with which such castings can be made.

2 When the casting of type was perfected, it saved so much time that its usefulness for making other pieces which were wanted in large quantities was at once evident. Accordingly many men have made efforts to perfect a machine or process by which they could turn out finished castings correct as to size and shape. Since type was first successfully cast it is natural that the same methods should be used for making other castings. So far no material superior to steel, the material used for type molds, has been found for making the matrices, or dies, as they are called. Brass can be and is used where the casting is very intermittent as in a linotype machine, but for continuous work and where accuracy and truth are required, steel is necessary.

3 The next consideration after the material for the dies is that for the castings. Table 1 gives a few facts concerning the materials

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

that are of interest in connection with this subject, whether for making dies or castings.

4 There are only a few common metals with a melting point sufficiently low to be cast in steel dies, because these do not have to be brought to their melting point to be spoiled. If improperly treated in making, the dies are very susceptible to injury in use. Fine cracks are sometimes made in hardening the dies, causing them to break when subjected to the constant change of temperature which occurs in their use. Thin places in dies are more apt to get burned than heavy parts. For most work hardened dies are superior to soft, but their cost is very much greater.

TABLE 1 DATA OF MATERIALS USED FOR DIES AND CASTINGS

Material		Melting Point		Strength Lb. per Sq. In.		Modulus of Elasticity
		Deg. Cent.	Deg. Fahr.	Tensile	Compressive	
Aluminum	Al	657	1214	15000	12000	11,000,000
Antimony	Sb	630	1166	.....		
Bismuth	Bi	266	511	6400		
Cadmium	Cd	322	612	.....		
Copper	Cu	1072	1951	30000		
Lead	Pb	326	618	2000		
Magnesium	Mg	633	1171	.....		
Tin	Sn	230	446	4000		
Zinc	Zn	415	779	5500		
Iron	Fe	1505	2742	52000	85000	28,000,000
Nickel	Ni	1451	2644	.....		
Steel		1427	2600	65000	.....	33,000,000

5 In general, the method of making a casting is as follows: A suitable die or mold is made in steel. This must be so arranged that when opened, the casting will either drop out or be easily ejected. The die should have the fewest possible partings. This is important because it is easier to hold the casting to correct size when the die is in a few pieces than when it is in many. Also, the fewer the partings, the fewer the fins that will be made on the castings. Fig. 1 shows a complicated casting.

6 Whenever cores are necessary, suitable means must be provided for withdrawing these without deforming the casting. The die must close properly, that is in correct alignment, and be held firmly because the pressure of the fluid metal has to be overcome. The die being held closed in the machine, metal is forced into the die, pressure being provided by a pump which acts directly on the metal, by air

pressure on top of the molten metal or by a head of molten metal. The metal is allowed only to set. The mold is then opened and the casting removed, the casting being left in the mold only as long as is absolutely necessary. If the die is too hot, it is then cooled and the operation repeated. The temperature of the die is important, and should be held constant because the expansion of the die will change the size of the casting. This operation is generally carried on in so-called hand-casting machines, that is, machines operated by manual labor.

7 Automatic machines are in use and have great advantages in producing uniform castings because the conditions do not vary. The

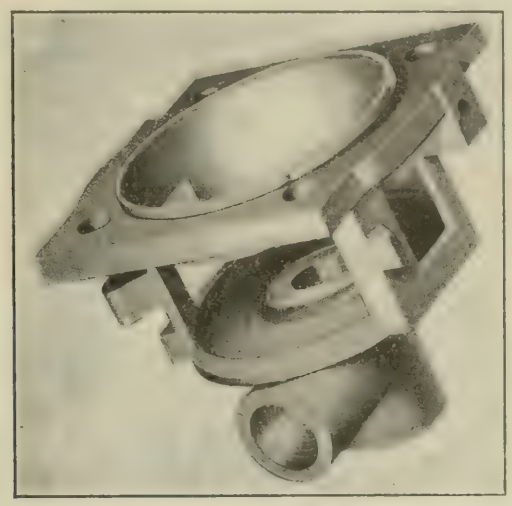


FIG. 1 A COMPLICATED CASTING

automatic machine is run at a constant speed and is not subject to varying conditions due to the operator becoming tired. Hand-casting machines produce quantities of 500 per day, more or less, depending on the size and complexity of the piece. Automatic machines produce the castings much more rapidly and save on labor just as does any other type of machinery.

8 While the writer has been able to find only one maker of casting machines who is willing to publish a description of them, a few illustrations are given to indicate the form of machine that is used, and it might be well to say that these actually show only the principles involved in each machine rather than the perfected mechanism.

9 Fig. 2 shows a machine in which the metal is pumped into the mold by hand. *A* is the hot, molten metal; *C* the charging chute; *H* a valve to close this; *E* the pump cylinder; *F* the plunger; *G*

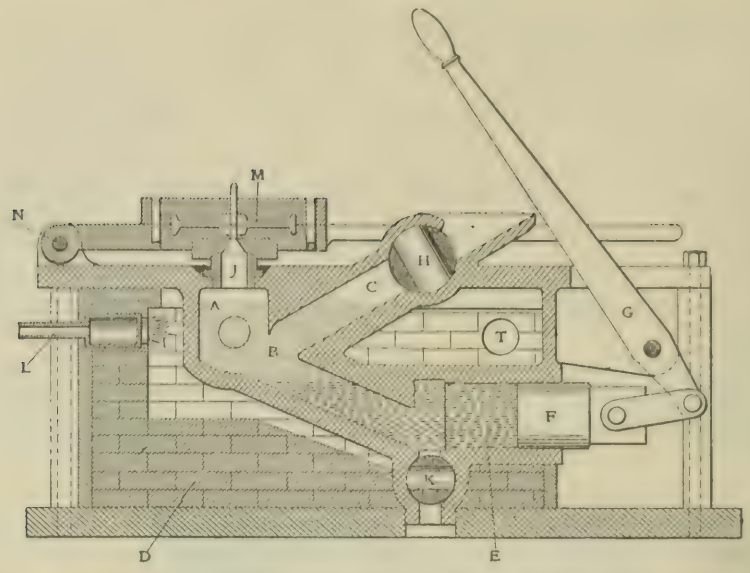


FIG. 2 MOLDING MACHINE WITH MOLD LYING FLAT ON TOP

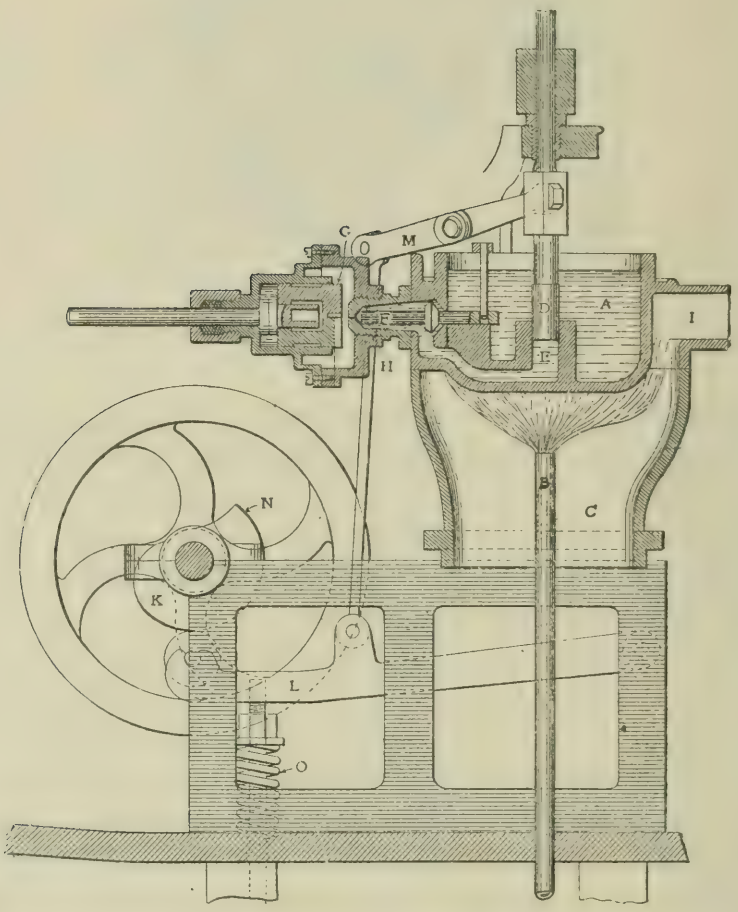


FIG. 3 AUTOMATIC MOLDING MACHINE

the pump lever; J the nozzle through which metal is forced into the mold M; K a valve for emptying the metal pot B. The gas for heat is supplied through pipe L, and the waste gases escape at T.

10 Fig. 3 shows an automatic machine. A is the molten metal; B the gas pipe; D the pump plunger; E the pump cylinder; F a valve to close the passage between the pump chamber and the metal pot, while metal is forced through the nozzle into the die or mold G which is brought up tightly against the nozzle while the metal is forced in. The left end of F closes the nozzle, while metal is drawn into the pump chamber by the raising of the plunger D.

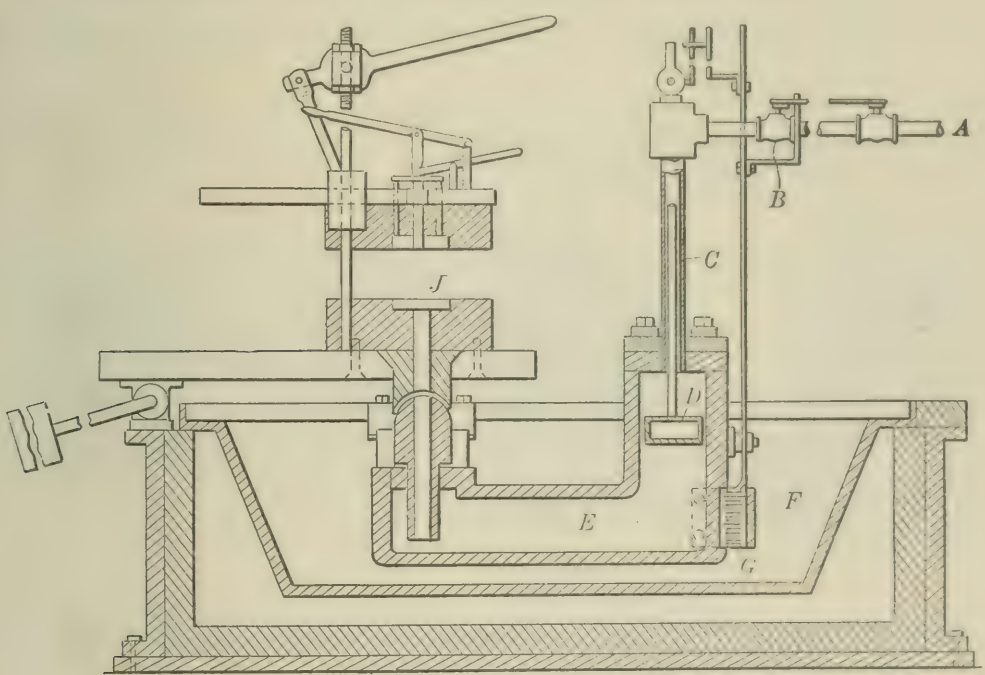


FIG. 4 MACHINE USING AIR PRESSURE TO FILL MOLD

11 Fig. 4 shows a hand machine in which air pressure is used to force the metal into the mold. Air enters at A and is controlled by the valve B. It comes down through the vertical pipe C and exerts a pressure on the loose piston D. This piston is hollow and floats on the metal contained in the pump chamber E. Metal is admitted to this chamber E from the main metal pot F through the valve G. The metal is forced by the pressure of the piston D out of the nozzle H into the mold J. The use of the piston D prevents oxidization of the metal from contact with the air which is admitted to force the metal into the mold. This air pressure is released before the valve G is opened so that metal can flow by gravity into the pump chamber.

The metal which is forced directly into the mold is taken from below the top surface of the molten metal and this precludes the oxidized skin on top of the metal from being forced into the dies. Any of this oxidized metal would make bad castings so that this point is important. The entire pump chamber in this arrangement is submerged in molten metal so that it is kept hot. The nozzle is so situated that it is probable extra means would have to be used to keep it heated and prevent the metal from freezing at this point. Undoubtedly in the practical working out of the machine this point was cared for.

12 The feature about die castings which most appeals to the manufacturer is their accuracy. So long as the dies are in good condition

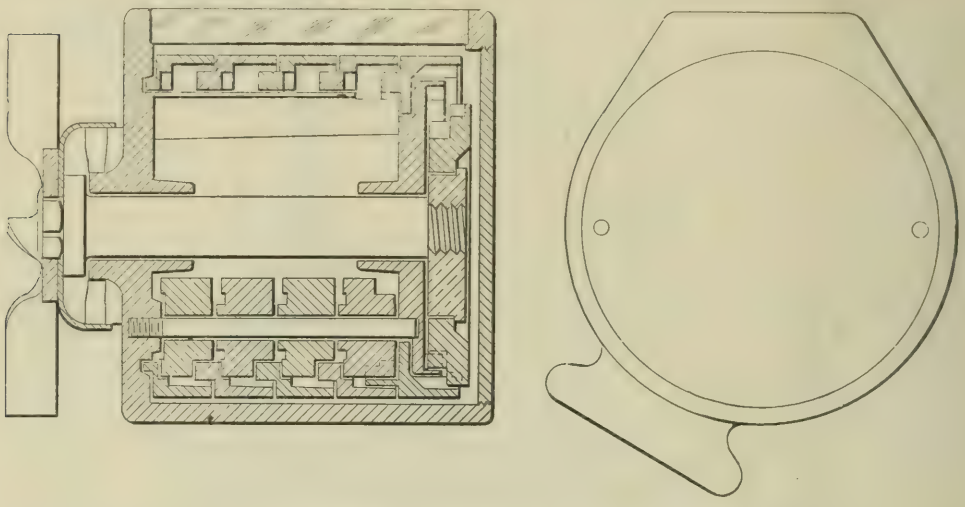


FIG. 5 VEEDER CYCLOMETER WITH SECTION VIEW

the castings should be perfectly true, both in size and shape. This does away with the necessity of gaging and makes it possible to use them without their having to be fitted into place by any hand work. A careful inspection is necessary, and this should be done as soon as possible after the castings are made so as to detect any flaws due to a slight failure of the dies. Any fins or burs formed by the necessary joints in the dies can be removed at the same time that the castings are inspected. Usually these fins are not thick and can be taken off thoroughly and quickly with a light hand tool.

13 A difficulty due to shrinkage is sometimes encountered when these castings are to be used in connection with parts made by some other process. While the shrinkage will be the same in two castings made in the same die if the metal is properly handled, it does not

always follow that it is proportionately the same when they are made from different molds. It is affected to a marked degree by the form of the casting, and if the castings are to be used in connection with machined pieces, it is advisable that these be machined to fit the casting. When it is not possible to do this and the casting must be fitted to existing pieces, sample castings should be made, and if these are not correct, the dies can be altered to make castings that are correct. In doing this, sufficient material must be allowed for grinding away, the casting being at first too small, although this makes a great deal of extra work.

14 The greatest gain obtains when an entire machine can be made of such castings. One of the best examples of this is the Veeder cyclometer so universally used on bicycles. This instrument is composed of about a dozen die castings and nearly as many parts of other materials. The number rings have 0.001 in. play, and yet these cast-

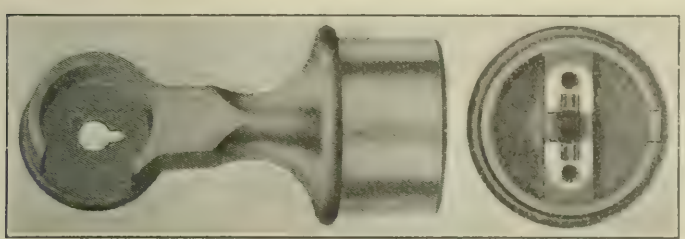


FIG. 6 HEAD FOR DESK TELEPHONE

FIG. 7 A LUG MATCHING FIG. 6

ings are made in enormous lots and, without being gaged, are assembled into the die-cast cases where they fit and work properly. So exact are these castings that when new, the instruments work more easily without lubrication than with it. This would obviously be possible only where the clearances are extremely small. A small dimension on these castings can be maintained to 0.0001 in. This instrument is illustrated in Fig. 5, the section view showing the rings both in their relation to the hub on which they turn and in contact with the transfer pinions. These latter turn on a German silver shaft 0.04 in. in diameter. The hub which forms the bearing for the number rings is so large in proportion to the width of these that they would tip and cramp if much end play were allowed. This makes it necessary to hold the dimensions on the rings correct within less than one-half of 0.001 in. In Figs. 6 and 7 are shown the head for a desk telephone, with a lug matching it.

15 The quality of these castings depends on both the alloy used and the method of making them. The alloys in most common use are those having a zinc base. This metal is useful in certain alloys, particularly with copper, but if alloyed with aluminum is unsuitable for most purposes. Aluminum-zinc alloys are frequently used because aluminum makes the alloy lighter in weight and in color. Some time after manufacture, castings of this alloy begin to disintegrate and will actually fall to pieces. In the disintegrating process they also change shape badly so that they should never, under any circumstances, be used for pieces which are wanted to stay flat or true. This alloy also tarnishes rapidly. Most of the alloys of zinc are too brittle to be of much use for the making of die castings. The usual effect of casting in metal molds is to make harder castings than would be made from the same metal in sand molds. For this reason a fairly ductile metal should be used.

16 The alloys of tin are being used quite extensively for die castings. These alloys possess the two desirable qualities of being quite ductile and of melting at a low temperature. Genuine babbitt metal, as it is commonly called, is an alloy consisting, according to the best authorities, of about 89 per cent tin, 7.3 per cent antimony and 3.7 per cent copper. This is a remarkably fine bearing metal and is possessed of sufficient strength to make it useful for many small parts. By certain changes in the alloy a composition even better for ordinary die-casting use can be produced.

17 The alloys of lead are used to a limited extent but, except for type, this metal is more often used in small proportions than as the foundation of the alloy. It is too heavy and lacks sufficient strength to make a very desirable metal. The percentage of lead in an alloy is sometimes increased for the sake of lowering the melting point and the price per pound.

18 Antimony is a common constituent of all die-cast alloys. The addition of this metal in the proper proportions, makes the alloy hard enough to machine well and increases the fluidity of the metal in its molten state. This makes it possible to obtain sharp castings such as are made from type metal, which is an alloy consisting of lead, $87\frac{1}{2}$ per cent, and antimony, $12\frac{1}{2}$ per cent.

19 In making an alloy of any two metals, when they are in correct proportions, there is formed a eutectic, the melting point of which is lower than that of either of the constituent metals. In some cases by the addition of a third metal, the melting point may be still further lowered.

20 The strength of die castings is of course largely dependent on the alloy, and also on the method of casting. Unless proper precautions are taken to insure sound castings, much trouble will be encountered from blow holes. To prevent these is often very difficult and sometimes impossible. They are caused both by the form of casting and by the tendency of the molten metal to hold minute bubbles of air or other gases which are carried into the mold and do

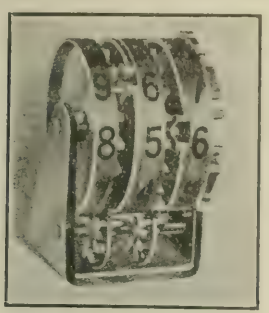


FIG. 8 SET OF NUMBER WHEELS AND PINIONS

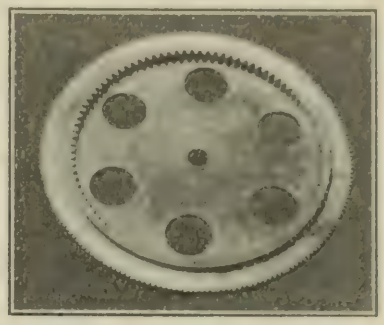


FIG. 9 INTERNAL AND EXTERNAL GEAR

FIG. 10 BEVEL GEAR

not escape before the casting sets. This is a case exactly parallel to that found in making iron castings. The shrinkage of the ordinary die casting is not as great as that of cast iron, and some of the tin alloys particularly shrink scarcely at all.

21 Another similarity to iron molding is found in the fact that the outside skin of the die casting is ordinarily stronger and harder than its interior. The hard outside skin is always desirable on a die casting, because the piece being cast to its finished size and no cutting being required, it is this outside skin which is first subjected to wear.

22 Exact figures on the strength of die castings might be quite misleading unless the exact composition and treatment of the castings were noted, so that general figures only can be given. They compare very favorably with cast iron in tensile strength, but are not nearly so strong in compression as that material. Their compressive strength is usually but little greater than the tensile, although it is somewhat, say one-third, greater. As the material is not as brittle as cast iron, it can be subjected to shocks which the iron would not

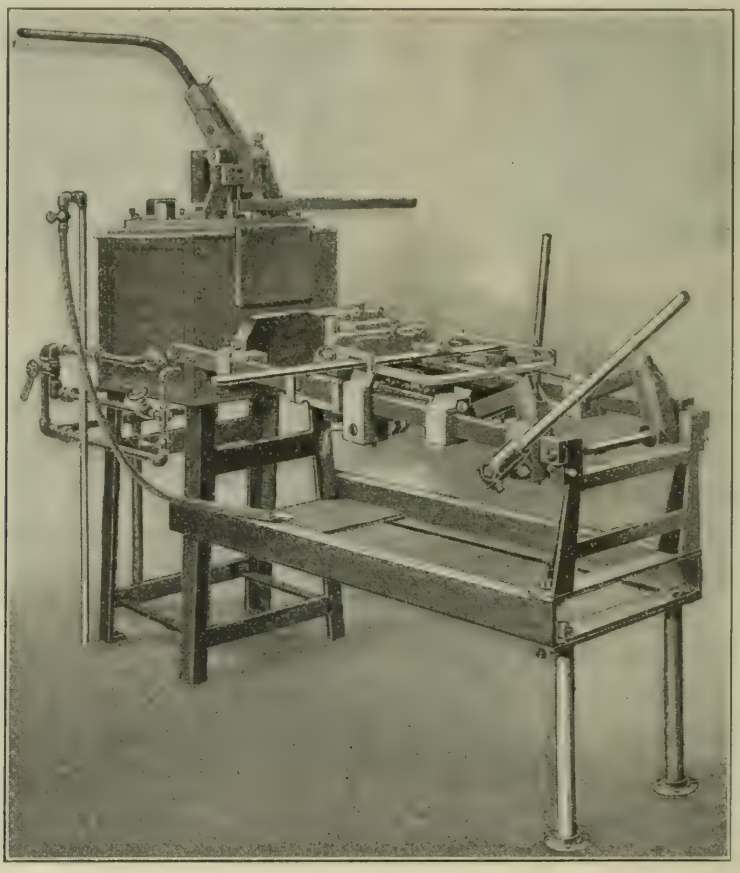


FIG. 11 SOSS DIE-CASTING MACHINE

withstand. Most die castings are much more ductile than cast iron. The greater ductility of the die castings is an advantage for threaded or tapped pieces. The castings do not chip out when tapped near an edge and a fine thread may be used. Some of the alloys are so ductile that the castings may be spun or riveted. This is often needed or is convenient for fastening two castings together or fastening a casting and some other piece.

23 Very little idea can be given of the cost of parts manufactured by the die-cast method. The tools or dies are costly or otherwise, according to their complexity or their simplicity. Practically all the die-casting manufacturers make the customer who ordered the castings pay for the dies. The parts are then charged for according to the time needed to cast them and the amount of metal used. As only the metal in the finished casting is charged for, there is no expense for scrap. The time for casting a complicated piece is little, if any, greater than for a simple one. From this it follows that the more complicated the casting, or the more difficult it would be to machine, the greater will be the saving effected by the use of die castings. The number of pieces to be made from a die is also pertinent, for hundreds of thousands of pieces can be made from some dies with no injury whatever to them. Other dies may need frequent repairs due to unavoidably weak points in their construction. Number wheels having internal or hooded gear teeth are good examples of parts which are profitably made by this process.

24 The set of number wheels and pinions shown in Fig. 8 will serve as a fair example for the comparison of costs by hand-casting machines and by automatic machines. The dies for this set of wheels and pinions for a hand machine would cost in the neighborhood of $450. The number-wheel castings would cost, in lots of 1000, about 10 cents each, and in lots of 10,000, about $8\frac{1}{2}$ cents each. The pinion castings in lots of 1000 would be 4 cents each and in lots of 10,000 would be 3 cents each. The cost of the dies for an automatic machine for the same wheels and pinions would be about $2000. The number-wheel castings in lots of 10,000 would cost about $2\frac{1}{2}$ cents each, and in lots of 25,000 or more about 2 cents each. The pinion castings would cost about $1\frac{1}{2}$ cents in 10,000 lots and in 25,000 or more lots, about 1 cent each.

25 The necessary die for the internal and external gear shown in Fig. 9 would cost about $100 for a hand-casting machine, and the castings about 13 cents each in 10,000 lots. The dies for the same piece to be used in an automatic machine would cost about $450, and the castings would cost about 2 cents each in 10,000 lots, plus 8 to 9 cents each for metal. The bevel gear illustrated in Fig. 10 was made in a hand-operated machine of a zinc alloy, commonly known as hard metal. A die for this costs about $90, and the casting $7\frac{1}{2}$ cents each in 1000 lots, and $6\frac{1}{2}$ cents each in 10,000 lots.

26 Fig. 11 shows the Soss die-casting machine. This machine is offered for sale and enables a manufacturer to make his own die castings with a comparatively inexpensive equipment.

No. 1335

VARIABLE-SPEED POWER TRANSMISSION

By George H. Barrus, Boston, Mass.
Member of the Society
and
Chas. M. Manly,[1] New York
Non-Member

A considerable range of speed can be conveniently obtained with direct current machinery, but neither alternating current, nor internal combustion or gas engines satisfy entirely the requirement of providing for a wide variation of speed with a large torque at low speed. The hydraulic variable-speed power-transmission mechanism, designed by C. M. Manly, is intended to provide a device which, when introduced between an engine and any device to be driven by it, not only gives a speed range from maximum in one direction through every intermediate speed down to zero, and then up to maximum in the reverse direction, but does so with the torque increasing in proportion to the decrease in speed, and permits of the most rapid and continuous reversal of the driven device without possibility of injuring the power transmission mechanism itself or the driving mechanism operating it.

2 The device, Fig. 1, consists of a pump connected to the power generator, with one or more fluid pressure motors attached to the driven shaft or shafts and placed in any desired location with reference to the pump, with hydraulic connections between them, and a single operating lever. The pump is of multi-cylinder construction with variable stroke, connected to the driving shaft. The multi-cylinder motors have a fixed stroke and are connected to the driven shaft, the pipe connections or passages between them transmitting the working fluid. The various cylinders, both of the pump and

[1] Vice-President, Manly Drive Co., 17 Battery Pl.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers. Published in abstract only. The complete paper appeared in The Journal, December 1911.

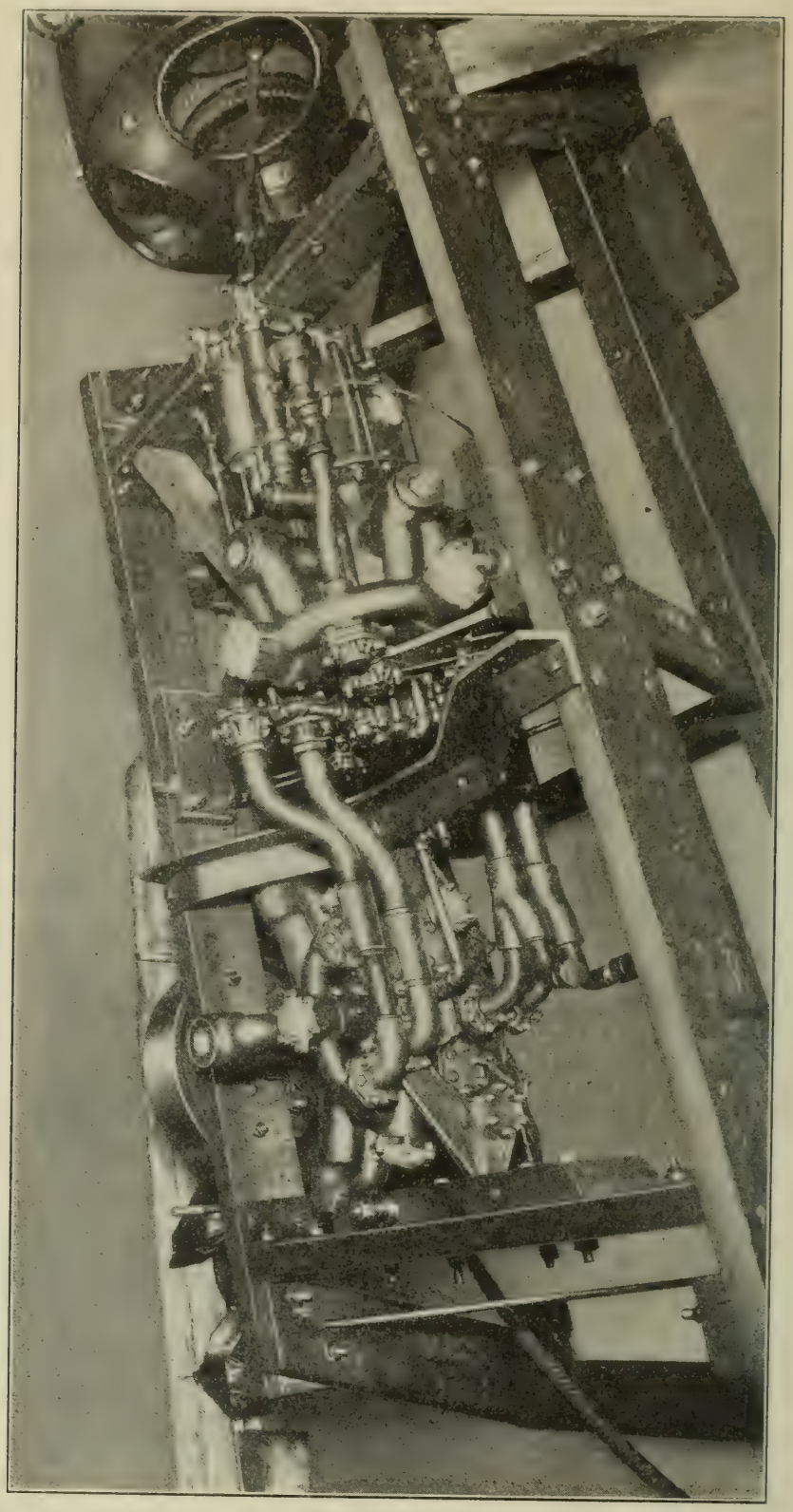


FIG. 1 GENERAL VIEW OF APPARATUS

motors, radiate equidistantly from a central crank chamber, and the pistons or plungers are connected to a single crank pin, which is common to all. The fluid used is ordinary machine oil, the lubricating qualities of which and its freedom from danger of freezing admirably fit it for such a purpose. When once filled, the oil is used over and over again, being in continuous circulation from pump to motor through one set of pipes or passages and back again from motor to pump through another set.

3 The stroke of the pump may be varied at will; that of the motor is fixed. When the pump is running at full stroke the motor operates at the highest speed. By varying the pump stroke and thereby the velocity of the oil in circulation, the motor runs at a speed which is exactly in proportion to the amount of oil that passes through it. Any desired rotative speed can therefore be secured and maintained. For reversal, the pump stroke first passes through the zero point, then the valves change and the oil is simply pumped in the opposite direction through the motors and pipes.

4 When the motor is going forward, Fig. 2, pipe E furnishes a supply to the motor transmitting the oil under pressure from pump to motor, while pipe F returns it from motor to pump, thereby answering the purpose of an exhaust pipe. When the motor goes backward, pipe F becomes the supply under pressure, and pipe E changes to the exhaust pipe, the direction of circulation through the connecting pipes being completely reversed. In case of a sudden check in the speed or a quick reversal, the momentum due to running in one direction is taken up in the device itself. A safety valve, set at 2000 lb. per sq. in., opens a by-pass when there is an over-pressure and this acts as a cushion preventing injury to the machine.

5 The center of the crank is half way between the center of the shaft and the center of the bushing when all three are in line. Therefore when the bushing is rotated 180 deg. around the center of the crank, the center of the bushing which is the real crank pin, is brought into exact alignment with the center of the shaft, and when this occurs, the length of the crank becomes zero, and the reciprocations cease.

6 The rotation of the crank bushing from the position of maximum stroke to the no-stroke point, is accomplished by the use of an auxiliary piston, lying parallel to the shaft, and supplied with power from the fluid pressure of the pump; this piston operates on the bushing through appropriate mechanism. It is under the control of a pilot valve, which is moved at will by means of a hand lever. By simply moving this lever from one end of its throw to a central posi-

tion, the speed of the motor shaft is varied from its maximum speed to a condition of absolute rest, and by moving the lever to the other end of its throw the motion is reversed and any speed is secured ranging from zero to a maximum speed in the reverse direction. Meanwhile the driving shaft continues to run at constant speed, whatever the speed or direction of motion of the driven shaft. When the motor comes to absolute rest, the pump stroke being reduced to zero, no motion of the motor is permissible in either direction until the adjustment is changed so that the fluid again begins to flow from pump to motor. Its effect at such time is that of a brake applied to the wheels,

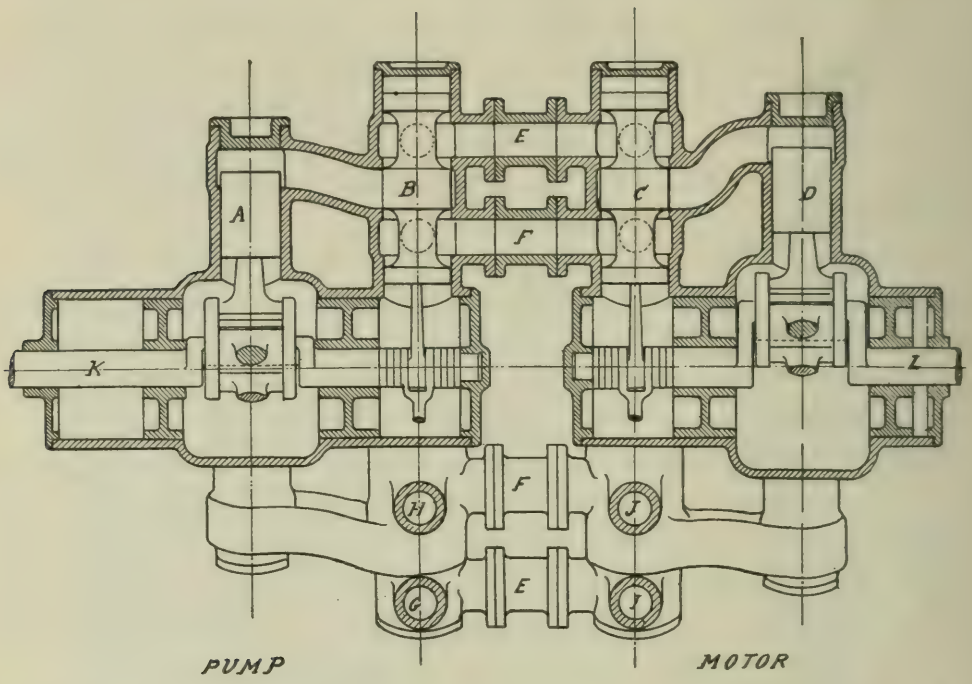


FIG. 2 SECTIONAL ELEVATION OF PUMP AND MOTOR

though much more positive and reliable. When the lever is in its forward position, the machine goes forward at maximum speed. When the lever is pulled over to the middle position, the machine comes to rest, and is locked there as with a brake. When the lever is pulled over still farther to its extreme backward position, the machine goes backward at maximum speed.

7 Four series of tests were made. The efficiency obtained on the tests of Series A, with medium temperature of oil and constant power, is 85.3 per cent at the full speed of 363 r.p.m., increasing to 89.4 per cent with speed reduced to 250 r.p.m. and then decreasing to 85.7

per cent with speed further reduced to 140 r.p.m. On the tests of Series B, with medium temperature of oil and constant weight on the brake arm, the efficiency is 86.2 per cent at the full speed of 352 r.p.m., increasing to 88.4 per cent with speed reduced to 264 r.p.m., and falling back to 83.1 per cent with speed decreased to 112 r.p.m. On the tests of Series C, with maximum temperature of oil and constant power, the efficiency is 87.4 per cent at full speed of 350 r.p.m., increasing to 91.9 per cent with speed reduced to 270 r.p.m., and then falling back to 87.0 per cent with a further reduction of speed to 163 r.p.m. On the tests of Series D, with maximum temperature of oil and constant weight on the brake arm, the efficiency is 87.4 per cent at the full speed of 350 r.p.m., increasing to 90.9 per cent with a reduction of speed to 245 r.p.m., and dropping to 81.6 per cent with a further reduction of speed to 105 r.p.m. The tests with maximum temperature of circulating oil show a material advantage over those with lower temperature as might be expected from the nature of the fluid. The average of all the figures of efficiency given computed numerically, is 87.7 per cent. There was never any sign of overheating.

No. 1336*a*

SYMPOSIUM ON OIL ENGINES

OIL ENGINES

By H. R. Setz,[1] Warren, Pa.

Non-Member

It has been reserved for modern engineering to give to liquid fuels the consideration due them by reason of their exceptional transportation and storage facilities, their high heating value and their small bulk as compared with all solid fuels.

2 The first successful solutions of the problem of using liquid fuels in internal-combustion engines were naturally attained with gasolene. This fuel, the lightest of the hydrocarbons obtained by fractional distillation of crude oil, vaporizes at comparatively low temperatures and readily forms an explosive mixture under ordinary atmospheric conditions. Only some form of carburetor was therefore required, aside from the usual constructive elements of the gas engine.

3 A far more serious problem presented itself when the utilization of the heavier hydrocarbons, such as kerosene, fuel oil and crude oil itself, was attempted. Many experiments showed that in order to get an explosive mixture, these heavy oils must first be converted at a comparatively high temperature into vapor before or during their mixture with air. For this purpose most oil engines have a hot chamber or vaporizer where the oil, after having been introduced as a liquid or in the form of spray, is converted into vapor and then taken up by and mixed with a current of air. The most notable methods now in use for securing a perfect mixture by means of a vaporizer are shown in Figs. 1, 2 and 3.

VAPORIZING AND MIXING

4 In Fig. 1 evaporation and mixing are effected during the compression stroke. The oil is injected into an incandescent hood or

[1] Chief Engineer, Struthers-Wells Co.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

chamber which, for starting, is heated up externally by means of a lamp, and afterwards kept red hot by the combustion of the mixture in it. During the compression stroke air from the cylinder rushes through the contracted opening into this chamber and mixes with the vapors therein, until at the end of the stroke, the right proportion of combustible to air is reached. The mixture is then ignited simply by direct contact with the hot walls of this vaporizing chamber, augmented slightly by the heat due to compression.

5 An arrangement commonly used by gasolene engine manufacturers to adapt their engines to the utilization of heavier hydrocarbons is shown in Fig. 2. The vaporizer chamber is provided with a jacket

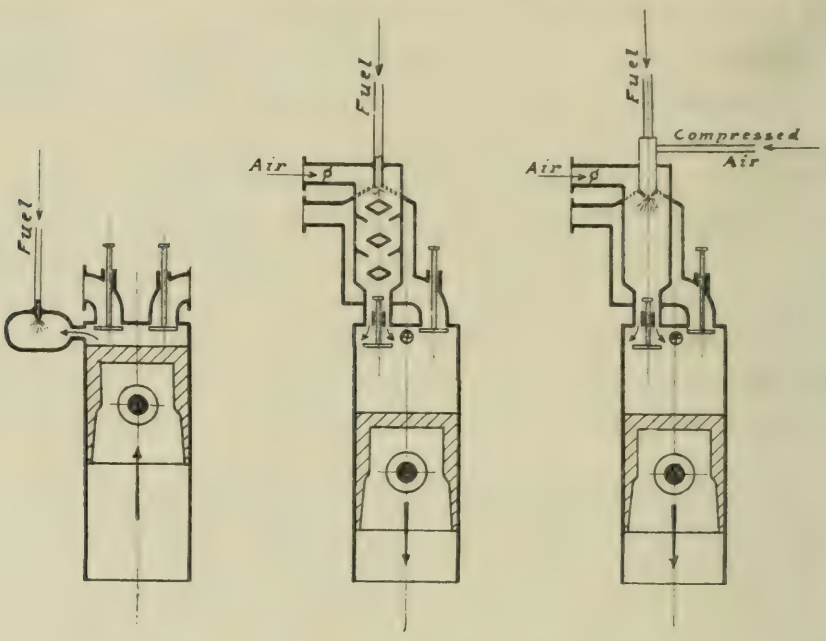


FIG. 1 FIG. 2 FIG. 3

METHODS FOR SECURING MIXTURE OF VAPOR AND AIR

space through which the exhaust gases pass, thus heating the vaporizer externally. A cloud of fuel vapor is produced by dropping the liquid fuel on the heated surfaces of the baffle plates inside the vaporizer. On the suction stroke of the piston free air enters this vaporizer and, in passing over the baffle plates, becomes heated and in the same time absorbs the oil vapors; the mixture thus formed and pre-heated then enters the cylinder and at the end of the compression stroke is ignited by an electric igniter.

6 In Fig. 3 the fuel oil is mixed with and broken up by a stream of compressed air of from 8 to 25 lb. pressure above atmosphere, so that

it enters the vaporizer chamber in the form of finely divided spray, and is immediately vaporized, due to the heat applied externally by the exhaust gases. The bulk of air, being aspirated during the suction stroke, then mixes with the fuel vapor and becomes preheated, thus forming the explosive charge. Compression and ignition are the same as in Fig. 2.

7 These types of oil engines, especially those of Figs. 1 and 2, are quite simple and therefore cheap in first cost. Their method of vaporization, however, is rather crude and gives rise to objections well borne out by practical experience, which are the cause of the prevailing prejudice against such oil engines. The chief drawback to all these vaporizers is the practical impossibility of vaporizing the fuel completely at all loads and under all conditions. The heat of the chamber should always be high enough to vaporize all the oil, but never hot enough to decompose it, or a deposit of carbon will be formed in the vaporizer and cylinder, accompanied by incomplete combustion, and therefore low efficiencies; this manifests itself by the objectionable smoke and odor of the exhaust gases. Another drawback is that in all engines of the type of Fig. 1, in order to obtain certainty of ignition and at the same time prevent pre-ignitions at different loads, the temperature of the vaporizer should vary with the load, a practical impossibility. The pre-heating of the mixture, as required for engines operating under the principles shown in Figs. 2 and 3, decreases the weight of the air aspirated, and therefore the capacity of the engine; while the throttling of the air in passing through the vaporizer chamber and passages, as well as the high back pressure due to the exhaust gases passing through the jacket space of the vaporizers, decreases the power output of such engines still more. The necessity of first heating the vaporizer externally by means of a lamp before the engine can be started is rather inconvenient as it takes at least five to ten minutes. The fuel consumption of these engines averages about 1 lb. of oil per b.h.p-hr., corresponding to a thermal efficiency of not over 15 per cent.

8 This outline covers in a general way the mechanical principles of some of the commercial liquid-fuel engines today on the market. A great variety of modifications is possible, such as the cycle of operation (2-stroke or 4-stroke), vertical or horizontal, regulation, valve gear, etc. However, whether these work on the 4-stroke or 2-stroke cycle, they all have in common the fact that the fuel and air mixture, after having been compressed, is instantaneously ignited, i.e., at constant volume, and according to this mode of heat application they

belong in the class of constant-volume engines. The efficiency of this thermodynamic cycle is greater with increasing degrees of compression; the latter, however, is limited on account of the danger of premature ignitions due to the compression temperature. Attempts have been made to attain high compression pressures at low temperatures by injecting water into the cylinder during the compression stroke. Banki of Hungary has succeeded in building a very efficient motor along these lines; he compressed as high as 250 lb. per sq. in., the resulting combustion pressures amounting to something like 700 lb. This necessitated engine parts of hydraulic press proportions which, together with a possibility of short-circuiting the ignition plugs on account of moisture deposits, failed to make this engine a commercial success. The use of a special scavenging pump in a 4-cycle engine was also attempted, by means of which fresh air could be swept through the cylinder at the end of the exhaust stroke, thus cleaning the clearance space from the residue of spent gases and at the same time cooling the combustion chamber walls. Although this permitted the compression pressure to be raised, the extra expense for the special scavenging pump did not warrant its adoption for general practice. Without such artificial means the safe limit of compression in the present constant-volume liquid-fuel engines has by long experience been found to be about 70 lb. in gasolene engines and hardly more than 60 lb. in kerosene engines with spontaneous (hot-bulb) ignition (Fig. 1). The efficiency of these types of engines is therefore not likely to be increased very much in the future. Considering this, as well as the undesirable features touched upon in the foregoing discussion, it is quite obvious that, as far as reliability and economy of operation are concerned, these engines fall far short of what may reasonably be expected, especially in view of what is being accomplished with gas engines.

DIESEL ENGINE TYPE

9 An engine of decided advantages over those just discussed is the Diesel type, having the following characteristics of operation: During the compression stroke the cylinder contains air only, which is being compressed to about 500 lb., the resulting temperature reaching a point sufficiently high to ignite any liquid fuel injected into it. At the end of the compression stroke, fuel is gradually injected by means of an air blast at a pressure about 250 to 500 lb. above the compression pressure in the cylinder. This high-pressure air blast

completely atomizes the fuel during the injection period, and carries its small particles directly into the highly compressed and heated air in the cylinder, where they are immediately vaporized and ignited. By this method combustion is effected without explosion, as the continued admission of the fuel to about 10 per cent of the expansion stroke causes the development of heat to take place at approximately constant pressure. Since the oil particles are burned immediately after their mixture with air, there is no possibility of deposits forming on the cylinder walls and combustion is so complete that the exhaust products are entirely smokeless and without odor. Numerous tests made on Diesel engines of different sizes show an average fuel consumption of less than ½ lb. of oil per h.p-hr., corresponding to a thermal efficiency of about 30 per cent.

10 To the particular feature of compressing air alone to such a pressure and temperature that it will immediately vaporize and ignite the fuel injected into it, the constant-pressure engine as embodied in the Diesel motor undoubtedly owes its success. It lends itself admirably to the utilization of liquid fuels, as it does away at once with carburetors or vaporizers, and igniters; moreover it allows the burning of any liquid fuel without special accessories. For this same reason it is singularly well adapted to operation on the 2-stroke cycle. The doubtful practice of scavenging the cylinder with the fuel and air mixture, with its attendant loss of mixture through the exhaust ports, and the possibility of backfires, is therefore entirely eliminated. The result is that it embodies the full benefit of the 2-stroke cycle, practically twice the power capacity of a 4-stroke cycle engine of the same cylinder dimensions and speed. Another point of equal importance is the fact that with decreasing loads the efficiency of the constant-pressure engine decreases but very little, while that of the constant-volume engine drops very rapidly with lighter loads.

11 In view of these points, therefore, there can hardly be any question that for the utilization of liquid fuels the constant-pressure engine is far superior to the constant-volume engine. The great number of Diesel engine installations working under greatly varying conditions and with all kinds of fuels, abundantly proves this, particularly in Europe, where during the last three years engines of this type to an aggregate of over 250,000 h.p. have been built. Considering the fact that at the beginning of this century the Diesel engine had hardly emerged from the experimental stage, this is truly a remarkable achievement which cannot fail to attract engineering activity towards further developments in so promising a field of enterprise.

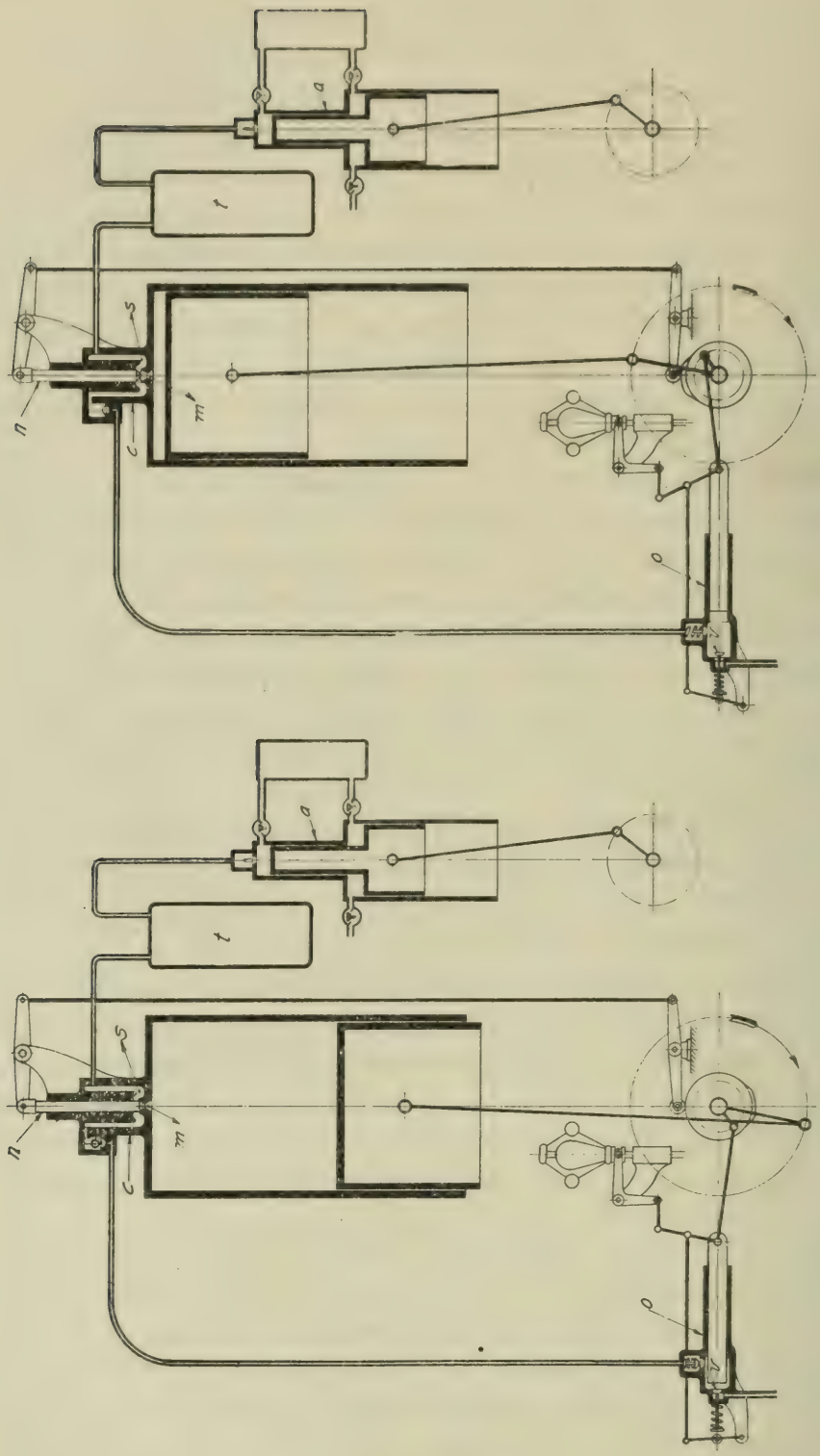


FIG. 4

DIESEL ENGINE WITH "CLOSED" INJECTION NOZZLE

FIG. 5

A number of new types of constant-pressure oil engines have been brought out within the last few years, all differing more or less from one another in their mode of fuel injection.

12 The accompanying illustrations show schematically the mechanical combinations used to inject the fuel and the principal modifications recently brought out. Since this new era of oil-engine construction started with the advent of the Diesel engine, we will first analyze the injection process of this engine, so as to establish the functions of the various parts entering into it.

13 Fig. 4 represents the period where a measured quantity of fuel, according to the load on the engine, is being deposited in space s of the injection valve cage c by the oil pump o, the injection valve n being closed at that moment. Space s is continuously in communication with the air storage tank t, into which the 2-stage air compressor a delivers the air required for fuel injection at a pressure of from 750 to 1000 lb. (One or two additional tanks are automatically kept charged by the compressor with air of about the same pressure for starting the engine.) The oil must therefore be delivered into space s against this high pressure which, in view of the small quantity to be delivered, requires extremely accurate work and adjustments on the oil pump o. Since fuel and injection air come into contact with each other while injection valve n is still closed, that is, before the actual injection period, it is quite obvious that the valve cage c, as well as the injection air, must be well cooled in order to prevent dangerous premature ignitions or the formation of deposits due to partial evaporation of the deposited fuel.

14 Fig. 5 shows the actual injection period, which starts as soon as injection valve n opens; the latter therefore controls simultaneously the admission of fuel and injection of air into the cylinder. At all loads the points of opening and closing of the injection valve n remain unchanged, i.e., the length of the period the injection valve is open is constant. Within this period a variable quantity of fuel, according to the load, is to be injected. In order to accomplish this most satisfactorily it has been found necessary to increase the pressure of the injection air with increasing loads on the engine, i.e., with increasing amounts of fuel to be injected; Diesel engine manufacturers recommend a pressure increase of about 250 lb. from light to maximum load. Since compression in the engine cylinder is constant at all loads, this is undoubtedly due to the fact that all the fuel having previously been deposited in space s, must be accelerated and atomized by the injection air as soon as injection valve n opens. Furthermore, if the

injection air pressure is too high at light loads it may happen that no ignition is effected on account of the cooling effect of the injection air, of which more will be said later on.

15 The variation of injection air pressures with varying loads on a 4-cylinder 250-h.p. engine is given in Table 1, which also shows the indicated compressor work. To the writer's knowledge no arrangements have so far been made on stationary engines to vary automatically the pressure of the injection air according to load variations; this must be done by hand, at the judgment of the engine operator.

TABLE 1 VARIATION OF INJECTION AIR PRESSURES WITH VARYING LOADS

ENGINE LOAD, B.H.P	INJECTION AIR PRESSURES, LB.	INJECTION AIR COMPRESSOR	
		I.h.p.	Per cent Engine Load
300	950	19.3	6.4
250	865	18.3	7.4
185	830	18.4	10.0
145	790	19.0	12.8

16 The foregoing analysis shows that the process of fuel injection resolves itself into two distinct phases, which are in no direct relation to each other:

a The measuring and depositing of the proper amount of fuel in the injection valve cage which is the function of the oil pump and may practically be performed at any time during the cycle of events in the engine. (Fig. 4 arbitrarily shows that this takes place at the beginning of the upward stroke of the piston.)

b The actual injection period, which is timed by the opening and closing of the injection valve, while it is the function of the injection air to pick up the fuel, atomize it and carry its small particles into the cylinder. The injection valve opens from 5 deg. to 8 deg. before the piston reaches the upper dead center, and closes about 28 deg. to 31 deg. past center. (The earlier opening and closing applies to heavier fuels.)

17 In order to properly distribute the oil and to direct the injection air so as to facilitate complete atomization, special accessories, atomizers or distributors, are used, which are placed in front of the injection nozzle *m*. Fig. 6 shows a typical atomizer used on

European Diesel engines. A series of plates *b*, arranged just below space *s* around the injection valve guide *g*, are provided with small holes in such a way that they straddle each other from plate to plate. These plates help to retain the oil after having been deposited in space *s*, while the holes will equally distribute it and mechanically divide the blast of injection air into small streams, thus disinte-

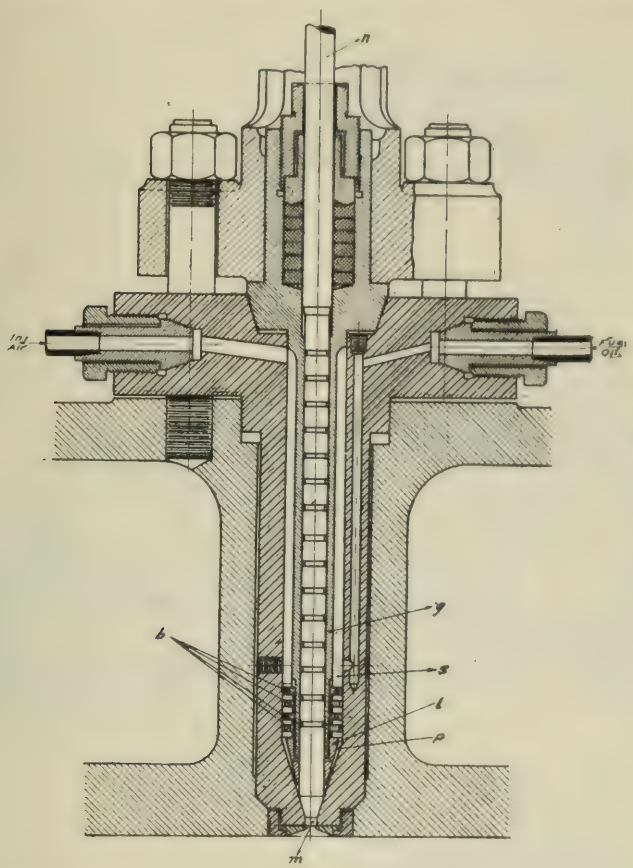


FIG 6. TYPICAL ATOMIZER USED ON EUROPEAN DIESEL ENGINES

grating the fuel passing down through them. By means of passages *p* arranged in the circumference of plug *l*, these streams are directed into the injection nozzle *m* where they acquire their maximum velocity. The resistance of the oil against the abrupt acceleration thus produced causes the oil to be disintegrated into small particles which are carried directly into the body of highly heated air in the combustion chamber.

AMERICAN DIESEL ENGINE

18 On the American Diesel engine, a cross-section of the latest design of which is shown in Fig. 7, the fuel injection valve and atomizer are arranged horizontally on the side of the combustion chamber. Owing to this horizontal position particular care must be taken to distribute the oil equally around the circumference of the injection

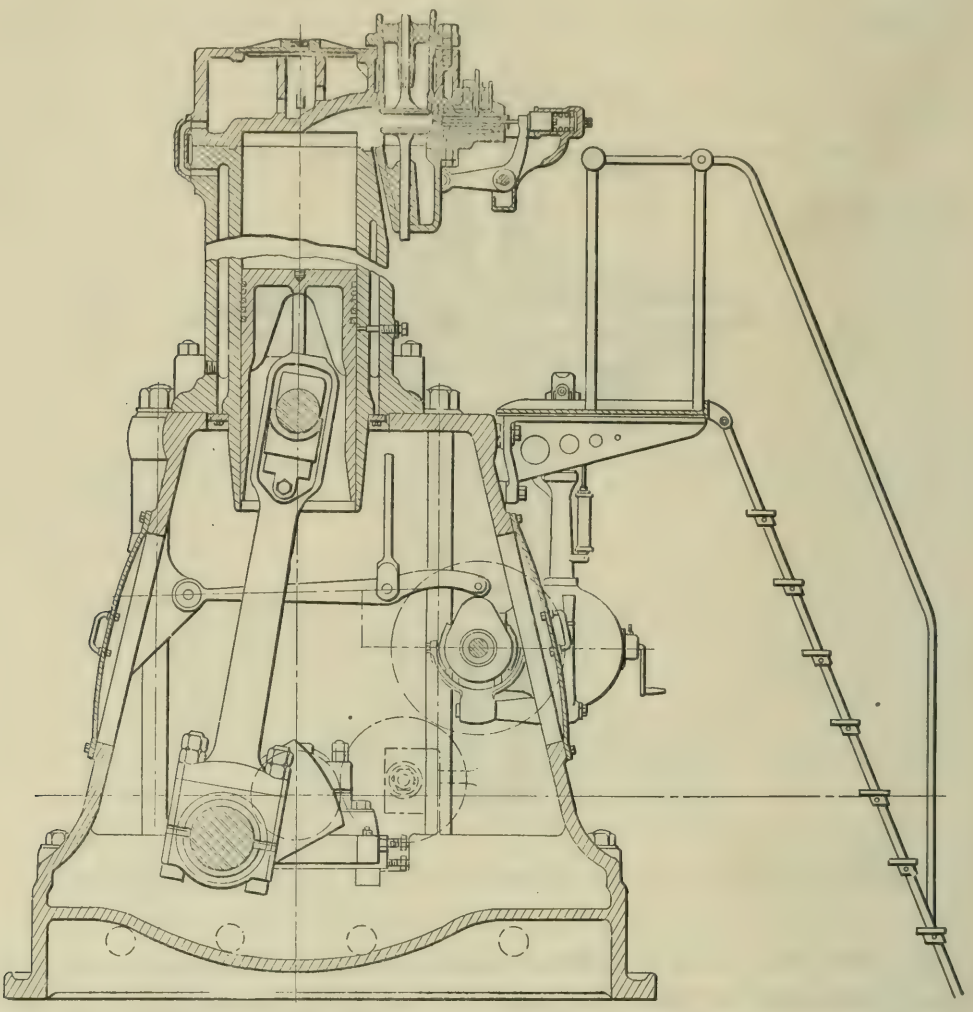


FIG. 7 CROSS-SECTION OF AMERICAN DIESEL ENGINE

valve. A sectional view of the atomizer is shown in Fig. 8. Oil and injection air come together in space *s*, the oil entering along passage *e* and annular ring space *r* through a ring of holes *h*. As the injection valve *n* opens, air and oil, being divided into small streams by a circle of holes *p*, are forced into the injection nozzle *m*, where these streams impinge upon each other, thus atomizing the fuel.

SABATHÉ ENGINE

19 A modification of the Diesel engine has recently been brought out in France, known as the Sabathé motor, which is evidently an attempt to eliminate the rather inconvenient requirement of variable injection air pressures with varying loads. Its fundamental features are identical with those of the Diesel engine (Figs. 4 and 5), with this exception, however, that not only the delivery of oil but also the lift of the injection valve n are here varied by the governor according to the load on the engine. Constructional details of the injection valve and nozzle are shown in Fig. 9. Aside from the injection valve n, a second valve v, sliding on n and being ordinarily held down on its

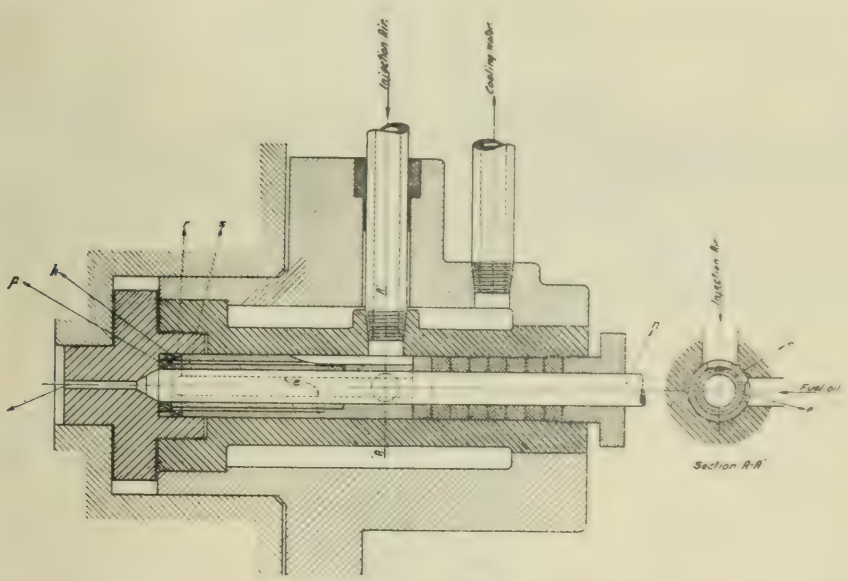


FIG. 8 ATOMIZER OF AMERICAN DIESEL ENGINE

seat by spring i, is provided. This valve v is lifted by collar r on the injection valve stem when the lift of the latter is sufficient to do so. On light loads only enough oil is delivered by the oil pump to fill chamber e underneath the valve v. This is blown into the cylinders when needle valve n lifts, the injection air passing down groove p in the needle valve stem. On heavier loads the amount of fuel delivered by the pump fills chamber e and overflows into space s; the lift of injection valve n is regulated by the governor in such a way that first the oil contained in chamber e is injected and then by lifting valve v, also that contained in space s. The pressure of the injection air is maintained at 800 lb.

DE LA VERGNE FH TYPE ENGINE

20 A notable combination of the Diesel principle, as illustrated by Figs. 4 and 5, with the hot-bulb arrangement of Fig. 1, is the type *FH* engine of the De La Vergne Machine Company, New York (Figs. 10 and 11). The engine operates on the 4-stroke cycle and compresses the air to about 250 to 300 lb., instead of 500 lb. as in the Diesel engine. The temperature thus obtained would not be high enough to

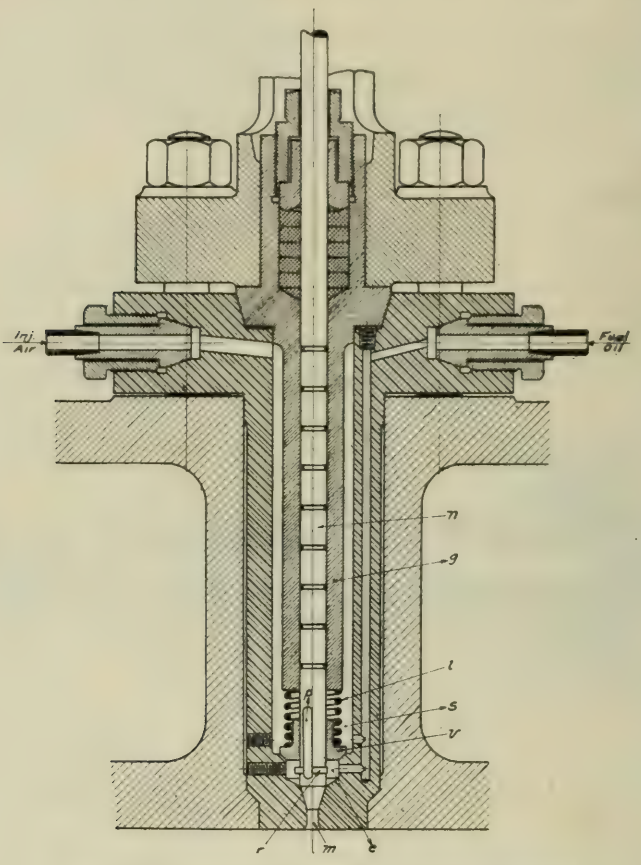


FIG. 9 ATOMIZER OF SABATHÉ ENGINE

ignite the fuel; recourse is therefore taken to a hot bulb *D*, the air in which, owing to the heat radiated from its uncooled walls, attains a higher temperature than that contained in the combustion chamber. For starting, this bulb is heated externally by means of a blow torch, 10 to 15 minutes being required for this purpose. At the end of the compression stroke the fuel is injected by means of an air blast of about 600 lb. pressure, from the injection nozzle *F* across the combustion chamber into the hot bulb *D*, where it is immediately ignited. Owing to the comparatively large distance the fuel spray has to tra-

verse after it leaves nozzle F until it is ignited in D, ignition first produces a considerable pressure increase (combustion at constant volume), to be followed by combustion at approximately constant pressure. This is shown on the two indicator cards (Fig. 12), where it will be noticed that the maximum pressures reach very nearly 500 lb., i.e., about the same as in the Diesel engine. As far as strains in the engine are concerned there is, therefore, not much difference between these two types. However, it must be remembered that in the Diesel engine this high pressure must be obtained at the end of the compression stroke in order to secure ignition, while in the De La Vergne engine, ignition is certain at about half that pressure. The requirements as to workmanship, and especially attendance, are therefore less severe.

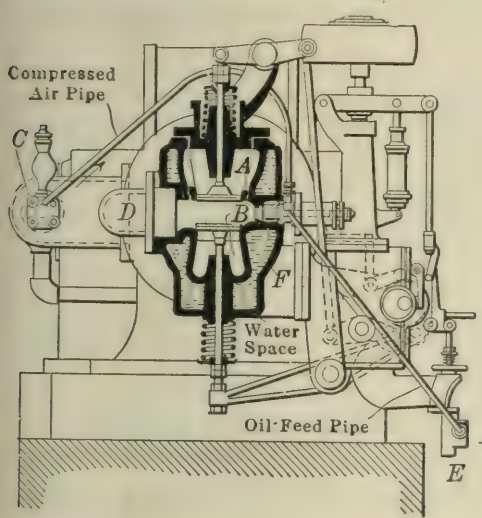


FIG. 10

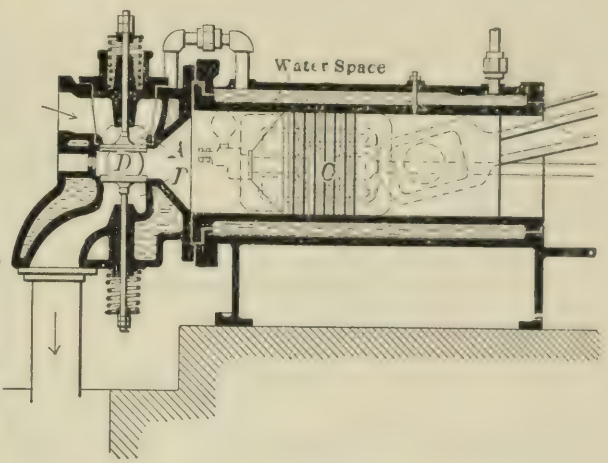


FIG. 11

TYPE *FH* DE LA VERGNE ENGINE

21 A 2-stage air compressor C, driven by an eccentric from the engine shaft, supplies the injection air. The air compressed by the first stage is stored in tanks t (Fig. 13), at a pressure of from 125 to 150 lb., and is available for starting the engine. The second stage of the compressor draws the air from one of these tanks, the amount drawn in being regulated by the governor by means of valve v to suit the varying charges of oil at each injection, and forces it directly into the injection valve cage, without an intermediary storage tank. It is needless to say that the resulting absence of tanks under extremely high pressures is a very desirable feature, while on the other hand it is, of course, imperative that the high-pressure stage of the compressor works with absolute certainty.

22 Fig. 14 shows constructional details of the injection valve and atomizer used on the type *FH* engine. Oil and injection air come together in annular space *s* formed between the injection valve guide *g* and cage *a*. As the injection valve *n* opens, oil and air proceed along the outside of guide *g* and are forced to pass through a series of chambers connected by a system of fine diagonal channels *d* on the outside of *g*. The oil is thus equally distributed around the circumference of needle valve guide *g* and enters injection nozzle *m* in a state of fine subdivision from where it is blown into the combustion chamber and hot bulb.

23 Undoubtedly owing to the fact that part of the oil charge is burned at constant volume, the fuel consumption of this engine is remarkably low considering the comparatively low compression. Thus according to tests made by Dr. Waldo on a 125-h.p. engine[1], the

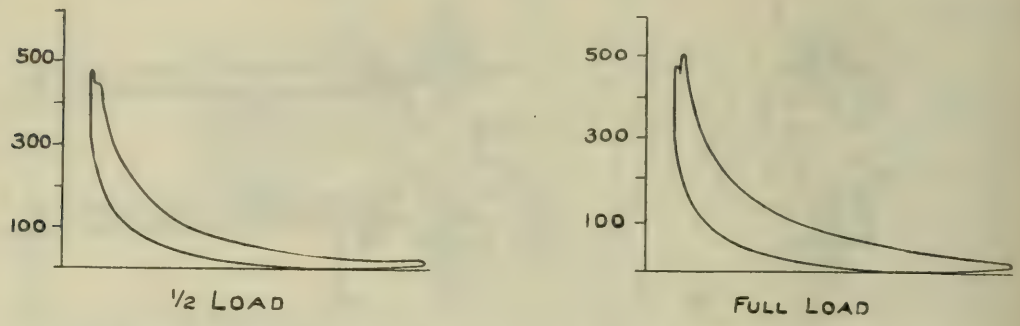


FIG. 12 INDICATOR CARDS, TYPE *FH* DE LA VERGNE ENGINE

minimum oil consumption was found to be 0.374 lb. per b.h.p-hr. with the engine carrying a load of 129 b.h.p. at 157 r.p.m. A report from the Snead and Company Iron Works, Jersey City, N. J., shows the following results for twelve months' operation. The plant, consisting of a twin cylinder 17 in. by 27½ in. engine, operated 3033 hr. at a load factor of 54 per cent; fuel oil consumption per h.p-hr., 0.506 lb.

OIL-PUMP ARRANGEMENT

24 The fact that the function of the fuel oil pump is in no direct relation to the cycle of events in the cylinder is taken advantage of by some European Diesel engine manufacturers on their multi-cylinder engines. Instead of using one oil pump for every cylinder, as is, for instance, the practice of the American Diesel Engine Com-

[1]Engineering News, January 13, 1910.

pany, they use only one pump for all cylinders. This pump delivers the oil into a distributer where, by means of a series of check valves and restricted passages which artificially increase the resistance against flow, it is thus equally divided into as many streams as there are cylinders. Considering the severe conditions under which it has to work on the Diesel engine, the advantage derived from the resulting reduction of oil pump parts is obvious.

25 As to details of construction of the oil pump, current practice seems to indicate a preference for a positively operated plunger (operated by an eccentric) rather than one operated by a cam where the stroke is varied according to the load by letting the governor shift

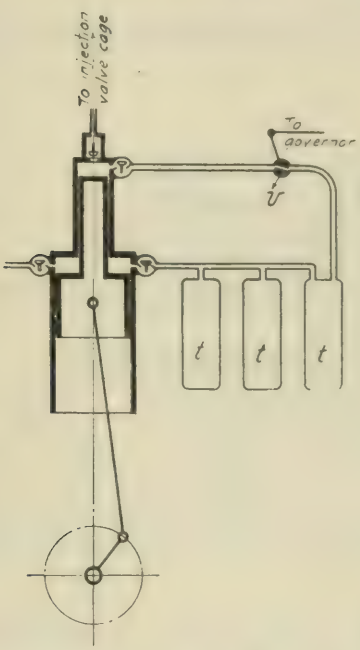


FIG. 13 SCHEME OF INJECTION AND STARTING AIR OUTFIT OF TYPE *FH* DE LA VERGNE ENGINE

a wedge block in between the cam and the plunger. In the former the displacement of the plunger is of course constant and considerably larger than that required for the maximum charge of fuel oil; the excess amount of oil is discharged through the suction valve *v* (Figs. 4 and 5), the opening and closing of which is determined by the position of the governor. This valve and also the mechanism for its operation are constructive elements which require the utmost care in design as well as workmanship, as upon their proper action depends primarily the accuracy of fuel oil delivery. Frictional resistance of the valve, which is liable to prevent its prompt closing, is

especially important and must be reduced to a minimum. It is therefore not advisable to have the valve stem pass through a stuffing box. The best practice at present is to use a positively-operated plunger for the operation of the inlet valve, this plunger passing through the stuffing box and thus practically eliminating friction as far as the valve is concerned.

MODIFICATION OF DIESEL TYPE WITH OPEN FUEL INJECTION NOZZLE

26 A very promising departure from the original Diesel engine which has been developed particularly for horizontal engines during the last two or three years by several German designers, is represented by Figs. 15 and 16. Its distinguishing feature is the open fuel injec-

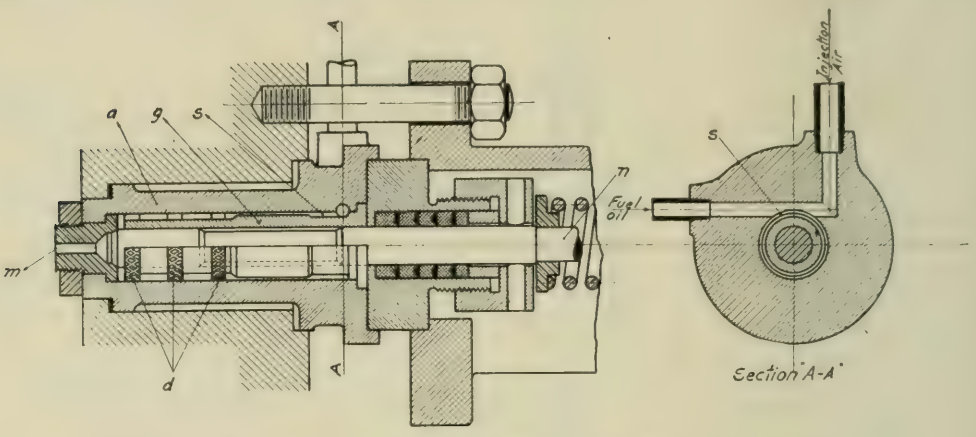


FIG. 14 ATOMIZER OF TYPE *FH* DE LA VERGNE ENGINE

tion nozzle *m*, through which space *s* is continuously in communication with the engine cylinder. Fig. 15 shows again the period where the oil pump *o* delivers a measured quantity of oil into space *s*; this happens when the piston begins its compression stroke, i.e., when the pressure in the engine cylinder and therefore in space *s* is low. Valve *n* opens shortly before the end of the compression stroke (Fig. 16), thereby admitting a blast of injection air, which picks up the fuel deposited in space *s* and blows it through injection nozzle *m* into the cylinder. Injection air of pressures varying with the load, the same as on the original Diesel engine, is taken from the storage tank *t* kept charged by the separate air compressor *a*.

27 Constructional details of a typical injection nozzle and air admission valve are shown in Figs. 17 and 18. It will be noticed that no atomizer is used; the oil is blown directly from space *s* through

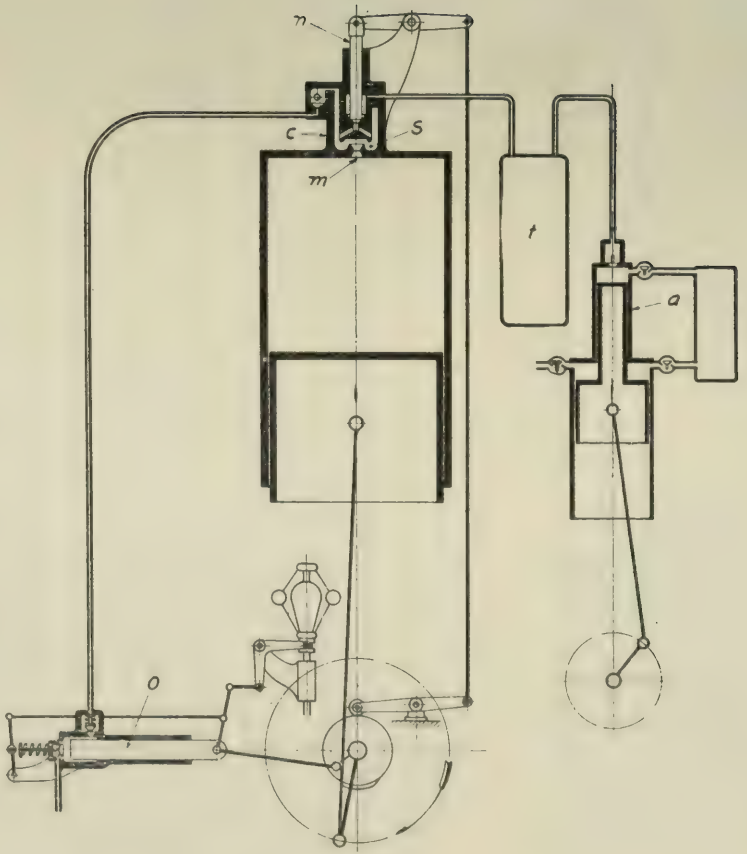


Fig 15

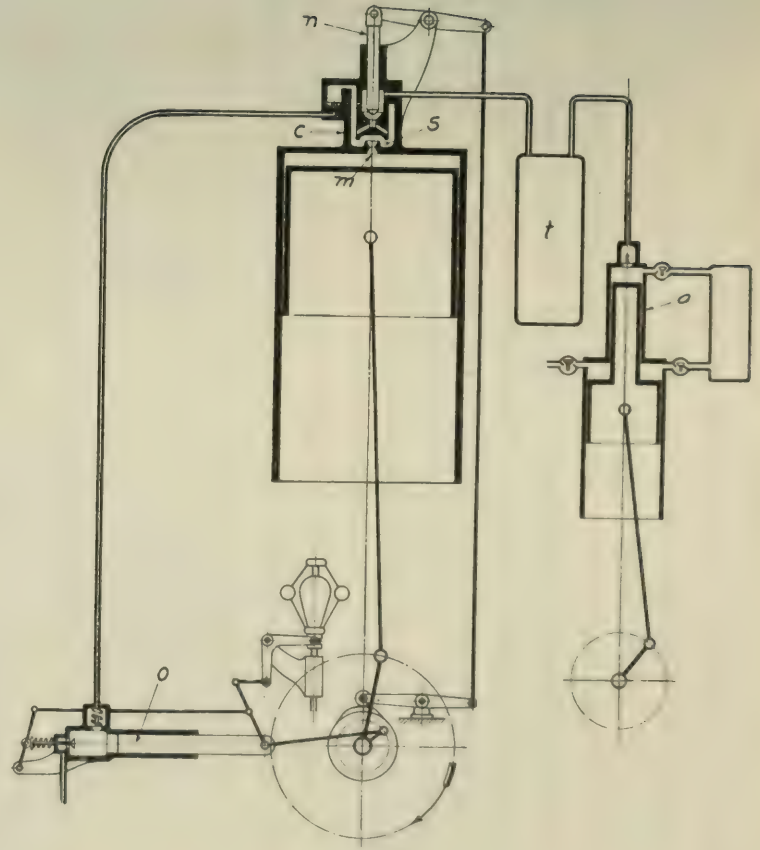


Fig. 16

Diesel Engine with "Open" Injection Nozzle

injection nozzle *m* into the cylinder. The air admission valve *n* is operated by a push rod *r*, provided with a valve *v* on its inner end which prevents any leakage along rod *r*, except during the very short interval where it opens the air admission valve *n*. A stuffing box, such as on the injection valves *n* of the original Diesel engine (Figs. 6 and 7), is therefore not necessary. At the heavy pressures under which these stuffing boxes have to work it happens quite easily that by excessive tightening of the glands, the valves are prevented from closing properly, thus causing loss of injection air and even premature ignitions. By means of a test cock *K* the proper delivery of oil may be ascertained.

28 As already mentioned air of approximately the same pressure as injection air is being used for starting Diesel engines. It is evident that under such enormous pressures a very small charge of air will suffice for this purpose. This is taken advantage of in this case by operating valve *n*, for starting, by the same cam and lever as when the engine is in actual operation. By opening by-pass valve *y* communication to the cylinder will be established through passage *p*, through which, in addition to the open injection nozzle *m*, enough air is admitted to start the engine. This makes an admirably compact and simple arrangement, although, on account of the short duration of the starting air admission period, the proper starting point of the engine must be closely observed.

29 The method of depositing fuel in the open space *s* is undoubtedly of advantage so far as the oil pump is concerned as it permits the delivery of oil against a far lower pressure than in the original Diesel engine.

AIR SUPPLY

30 In all the various types of modern oil engines just discussed the apparatus required to obtain the high-pressure injection air forms a comparatively complicated and therefore expensive accessory. The 2-stage air compressor contains four valves. Either the suction valve on the low-pressure or high-pressure stage must be provided with an adjusting device to vary the amount of air drawn in according to the oil charge. The storage tank *t* should also be provided with three valves, one to close it off towards the engine, one towards the compressor, and one safety valve. All these parts, together with a pipe line of sometimes considerable length with many joints, being continuously under a pressure of from 750 to 1000 lb., require no small degree of attention. That on the smaller units particularly, where

on account of their smallness these parts become extremely delicate, this becomes a matter of considerable importance, may be judged from the following dimensions of the 2-stage air compressor of the latest 5-h.p. Diesel engine: low-pressure cylinder diameter $2\frac{3}{4}$ in.; high-pressure cylinder diameter $\frac{15}{16}$ in.; stroke $2\frac{3}{8}$ in.

31 Attempts have been made to simplify matters by attaching one single-stage compressor, driven by a lever from the connecting rod, to each cylinder, this compressor receiving precompressed air out of the cylinder at a pressure of from 100 to 150 lb. This, however, necessitated an extra valve in each cylinder head, aside from other con-

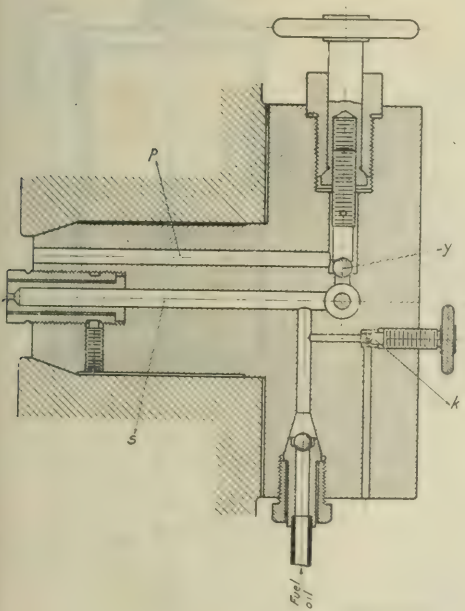


FIG. 17

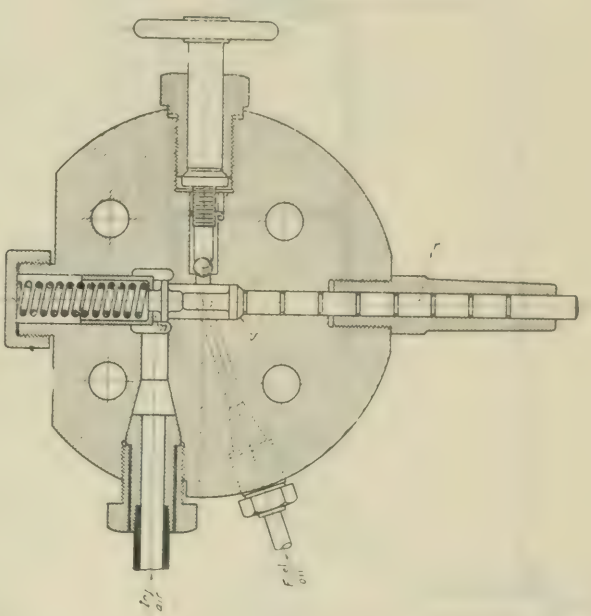


FIG. 18

TYPICAL "OPEN" INJECTION NOZZLE AND AIR VALVE

structive complications, and it was found furthermore that oil vapors, taken over with the air from the engine cylinder, had a tendency to foul the compressor valves, thereby rendering proper compression impossible.

32 In many respects far more promising at least for smaller units is the scheme to eliminate the air compressor entirely and instead to generate the injection air right in the engine at the moment fuel injection is to take place (Figs. 19 and 20). The special 2-stage air compressor is here replaced by a small chamber a, forming part of the engine cylinder and communicating with the latter through passages i and k. During the charging period (Fig. 19), the auxiliary piston

t is in its lowest position, so that port i is wide open, while the oil pump o delivers the proper amount of fuel into space s which communicates with the engine cylinder through the open injection nozzle m. Auxiliary piston t remains in this position until almost to the end of the compression stroke; up to this time the air pressure in chamber a will therefore be the same as in the engine cylinder.

33 To start the fuel injection (Fig. 20), auxiliary piston t is caused to move quickly upwards, thereby first covering passage i, and then

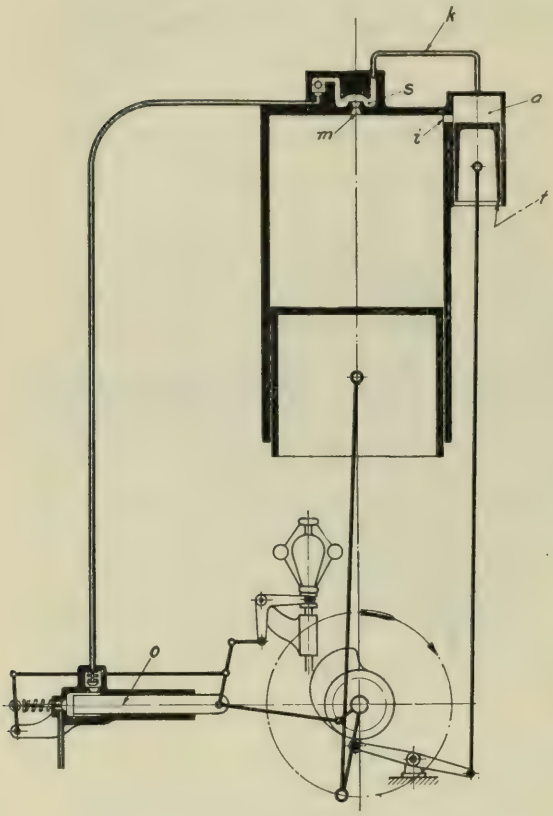


Fig. 19

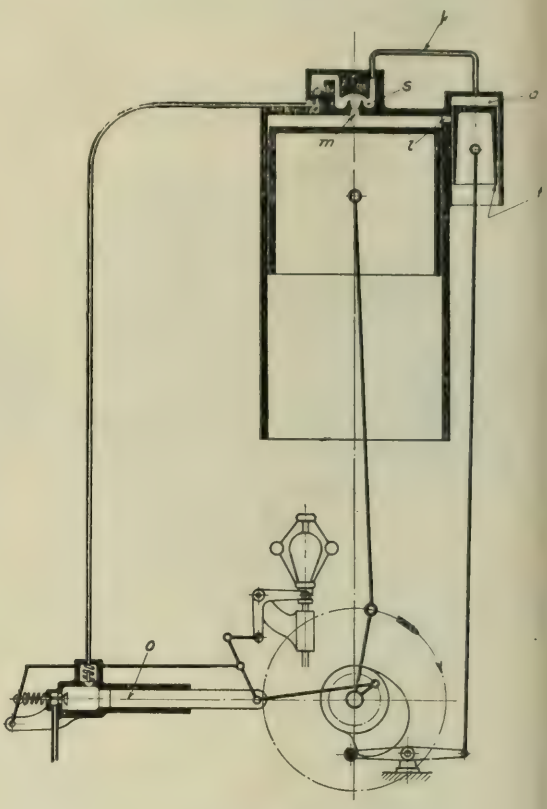


Fig. 20

Oil Engine with Auxiliary Injection Air Piston

compressing the air in chamber a to a pressure higher than the one in the cylinder. This causes the air to flow from a through passage k into space s where it picks up the deposited fuel and blows it through the injection nozzle m into the cylinder.

TRINKLER-KÖRTING ENGINE

34 Fig. 21 shows a cut through the head end of an engine of this type, designed by Trinkler-Körting and built to work on the 4-stroke

cycle. The auxiliary piston *t* is provided with a stem of comparatively large diameter whereby the effective piston area is greatly reduced on this end; the piston stem passes through a stuffing box to the outside where it engages at *c* with lever *cde*. This lever, together with bell crank *fgh*, form a releasing gear operated by cams *b*. As shown on the cut, lever *cde* is just being released at *h;* owing to the larger

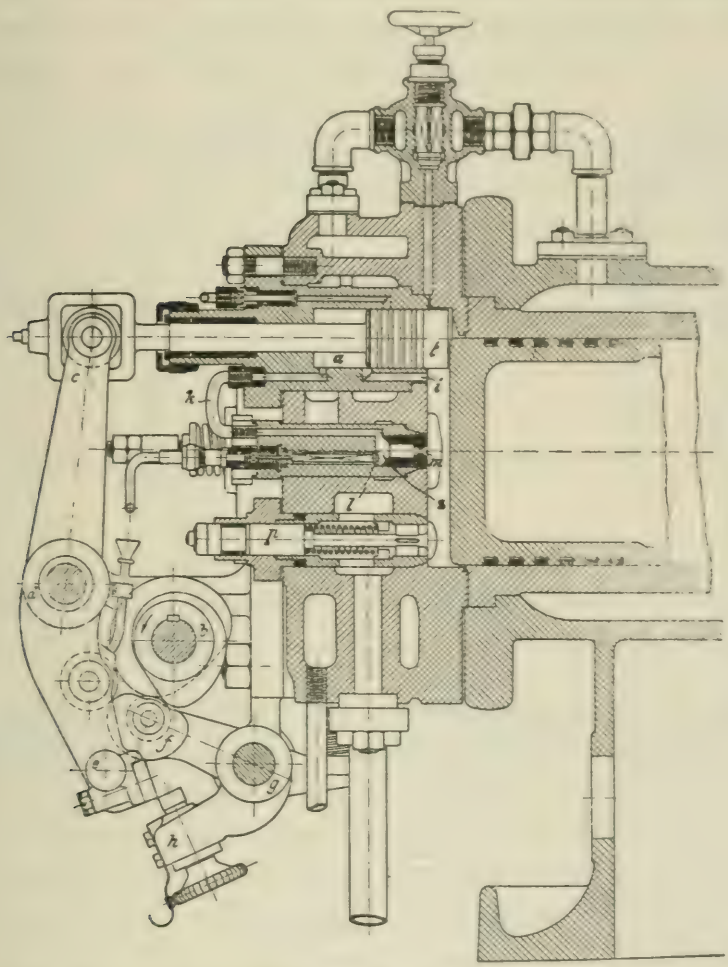


FIG. 21 SECTION THROUGH HEAD, TRINKLER-KÖRTING ENGINE

total pressure acting on the full piston area, this piston *t* is now caused to move quickly outwards, with the result already explained.

35 From the foregoing it will be noticed that while injection air is being generated in the chamber *a*, fuel is already deposited in space *s*. Since the latter is in communication with *a*, through the open passage *k* on one side, and with the engine cylinder through the open injection nozzle *m* on the other side, it is quite obvious that auxiliary piston *t* must be made to move at the highest possible speed so as to

prevent as much as possible the injection of fuel before the injection air has acquired the velocity necessary to atomize the fuel. This explains the use of a releasing gear for the auxiliary piston as shown in Fig. 21, although its limitations are well known; even on speeds which are considered moderate on internal-combustion engines this gear is liable to work rather noisily.

36 By depositing the fuel into the open space s at the beginning of the compression stroke the same advantages are derived as far as the oil pump is concerned as on the modified Diesel engine. A less desirable feature is the fact that the oil has to lie during the whole compression stroke in space s in contact with surfaces and air which attain high temperatures; this facilitates partial evaporation of the fuel, premature ignition if the fuel contains components of low volatility, and especially the formation of deposits which require frequent cleaning of the nozzle. This criticism applies to a certain

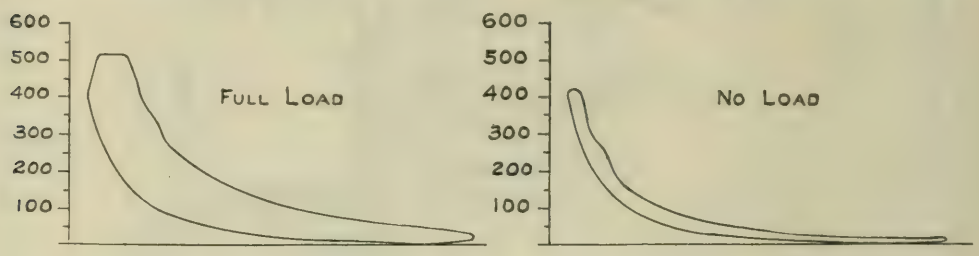


Fig 22 Indicator Cards, Trinkler-Körting Engine

extent also to the modified Diesel engine (Figs. 15 and 16), although it must be remembered that there the injection air, having been well cooled in the 2-stage compressor a and in tank t, helps to cool space s during the injection period, while in the Trinkler engine the temperature of the injection air will help to keep the temperature in s high.

37 A matter of considerable difficulty seems to be to pack properly the auxiliary piston stem. Since the auxiliary piston is exposed on both ends to the highest temperatures, while there is practically no means for cooling it nor even its chamber a, the temperature of its stem becomes considerably higher than what is considered allowable in stuffing box practice. In addition to this the stuffing box should be able to withstand maximum pressures of about 600 lb. All leakage along the stem should of course be prevented, especially since the air available for fuel injection amounts to only about one-tenth the clearance volume in the engine cylinder. On the other hand if the stuffing box gland is tightened too much, friction easily becomes so excessive as to render inaccurate the movement of piston t, which alone times the fuel injection period in this engine.

38 The pressure of the injection air generated in this engine remains practically constant at all loads, since it depends entirely upon the difference of the effective areas of piston t, i.e., the diameter of the piston stem. A variation of injection air pressures to suit the varying amounts of fuel, such as is found to be of advantage in the Diesel engine, is therefore out of the question.

39 Fig. 22 shows two indicator cards taken on a Trinkler engine.[1] It will be noticed that the compression pressure is 400 lb. At normal

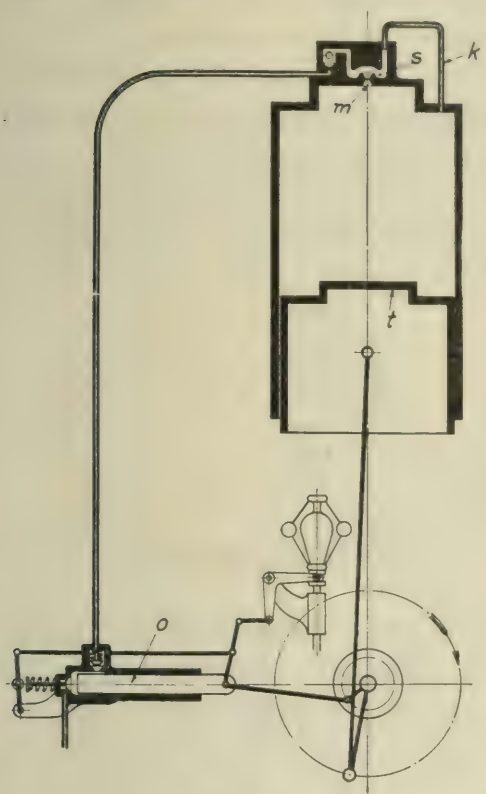


FIG. 23

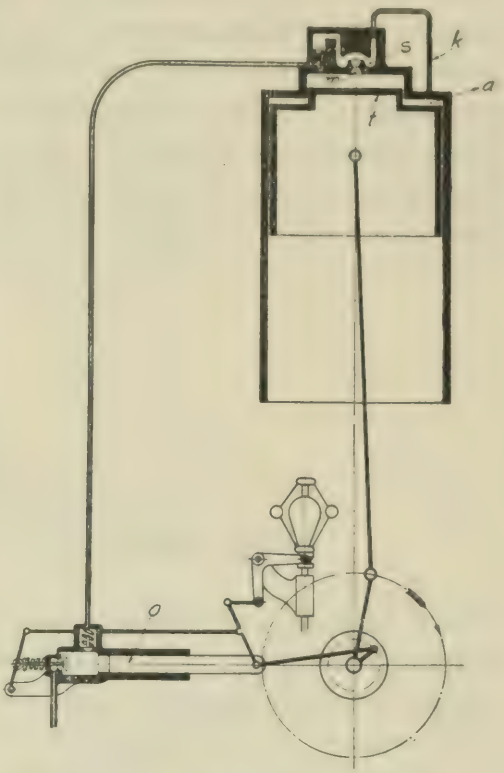


FIG. 24

HASELWANDER ENGINE

load combustion sets in, first at approximately constant volume and then at a constant pressure of about 500 lb. On a 12-h.p. motor the consumption of Russian fuel oil was found to be 0.48 lb. per b.h.p. at normal load, and 0.52 lb. per b.h.p. at one-half load.

HASELWANDER ENGINE

40 Considerably simpler than the Trinkler engine although not working under quite the same principle, is the Haselwander engine,

[1] Zeitschrift des Vereins deutscher Ingenieure, 1907, p. 903.

(Figs. 23 and 24). The auxiliary piston t of the former is here replaced by the projection t on top of the main piston, while the cylinder top carries a corresponding contraction. As in the previous two examples, fuel is delivered by the oil pump o into space s at the beginning of the compression stroke (Fig. 23), and remains stored there, in front of the open injection nozzle m, until near the end of the stroke.

41 When nearing the upper dead center projection t of the main piston enters the contracted cylinder top, thereby forming an annular chamber a in the cylinder (Fig. 24), in which the air is being compressed to a higher pressure than the body of air above t. The injection air thus obtained in annular chamber a now flows through k into space s, where it displaces the stored fuel and injects it through nozzle m into the cylinder. The height of projection t is, of course, limited by the desideratum that fuel injection and ignition must not take place until the piston has practically finished its compression

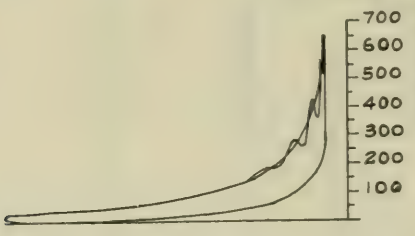


FIG. 25 INDICATOR CARD, HASELWANDER ENGINE

stroke, as otherwise the strain on the crank mechanism due to the sudden increase in pressure becomes excessive, especially after the engine is warmed up and ignition takes place much more quickly.

42 Strictly speaking the Haselwander engine does not therefore belong in the class of constant-pressure engines; fuel being injected during the short interval where projection t enters the contracted cylinder top and the end of the stroke, combustion will practically take place at constant volume. Since the air in the cylinder is compressed sufficiently high to ignite the fuel when injected into it, it follows that the maximum pressures in this engine must become rather high. This may be seen from the indicator card, Fig. 25, where the compression pressure is 240 lb., while the maximum explosion pressure amounts to about 650 lb. In his later designs, Haselwander has increased the compression pressure sufficiently to do away with the igniter which in his older models with lower compression was necessary for starting the cold engine. As far as thermal efficiency is concerned, this is a point in favor of the Haselwander engine, for it is well known that,

compression being equal, the constant-volume engine is superior to the constant-pressure engine. Tests made on a 10-h.p. single-cylinder, horizontal, 4-cycle engine of the older type, running at 250 r.p.m., gave the following results:

Load in b.h.p.	11.6	10.5	9.2	7.2	5.2
Fuel consumption per b.h.p-hr. in lb.	0.51	0.48	0.51	0.57	0.60

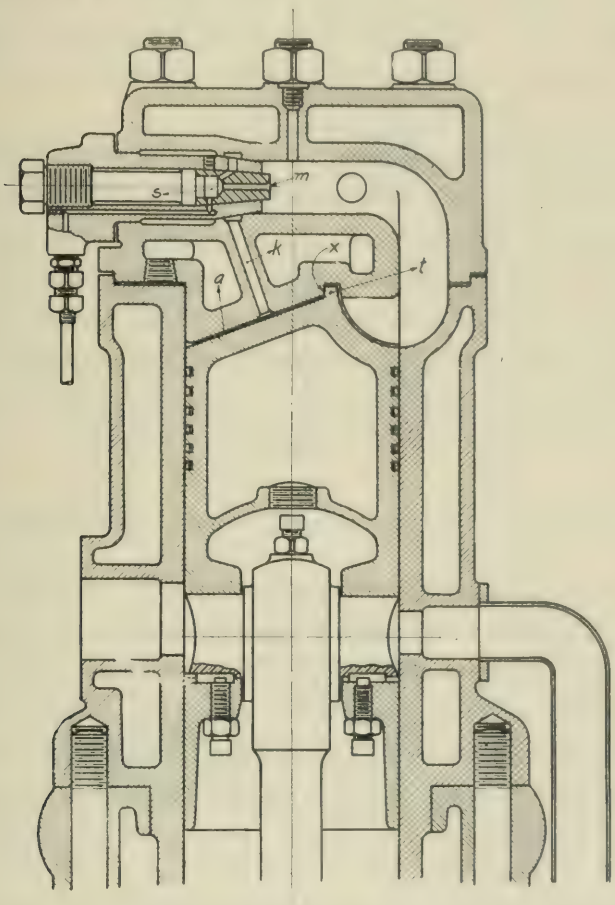


FIG. 26 HASELWANDER TWO-CYCLE ENGINE

The fuel used was Pechelbronner crude oil, specific gravity 0.814, heat value 18,350 b.t.u.

43 In order to prevent any possible ill effects from wear on piston and cylinder walls, the projection t fits into the contracted cylinder top with considerable clearance. This, however, produces the rather serious drawback of allowing the hot gases of combustion to flow past projection t into the annular space a, where the pressure is of course much lower at the time of explosion than in the combustion chamber.

As a result projection t will gradually be scorched and burnt off, thus increasing the clearance space to an extent such as to cause excessive leakage losses of injection air.

44 This difficulty, it is claimed by the inventor, has been entirely overcome by the design shown in Fig. 26. Here the projection t has been very cleverly combined with the baffle plate commonly used on top of the piston of ordinary 2-cycle engines. By laying this baffle t in the plane in which the side thrust of the connecting rod manifests itself, the influence of wear on piston and cylinder walls has been eliminated, making it possible to reduce the clearance at x to a minimum.

45 The drawbacks incidental to the method of depositing fuel at the beginning of the compression stroke in space s, in front of the open injection nozzle m, undoubtedly make themselves felt in a larger measure in this engine. Not only is the temperature in space s kept high by the hot injection air, but it is liable to be increased very considerably by the burning products of combustion being blown back from the combustion chamber through nozzle m and space s into a where the pressure is at that moment lower.

DISCUSSION OF ENGINE FEATURES OF DIFFERENT TYPES

46 The very favorable thermal efficiency of the Trinkler and Haselwander engines, their fuel consumption being almost equal to that of the Diesel engine, although obtained with far simpler means than in the latter, seems to point a way for future developments. Especially for units up to about 15 to 20 h.p., which by making proper provisions can be easily started by hand, the elimination of the extra air compressor is of great advantage. But even for larger units, where an air starting outfit is needed and as we have seen generally combined with the injection air compressor by Diesel engine manufacturers, the production of injection air by the principle incorporated in the Trinkler and Haselwander engines seems to offer certain advantages for the reason, as already mentioned in discussing the Diesel engine, that care must be taken to have the injection air well cooled; its temperature is very little higher than the engine-room temperature when it reaches the injection valve cage. The drop of pressure of the injection air during the injection period varies, according to the load, from 250 to 500 lb.; this, of course, is accompanied by a temperature drop. Since the volume of free air expended for every injection is equal to about one-twelfth to one-eleventh the displacement

of the piston, the amount of heat absorbed is therefore considerable. Thus by careful calorimetric measurements the operating temperature of the atomizer was determined to be 180 deg. fahr., which, considering the fact that its lower end reaches directly into the combustion chamber, is quite low. As a result the air and oil spray is cold enough when injected to lower materially the temperature of the air in the combustion chamber, thus endangering the certainty of ignition. This cooling effect manifested itself very strikingly

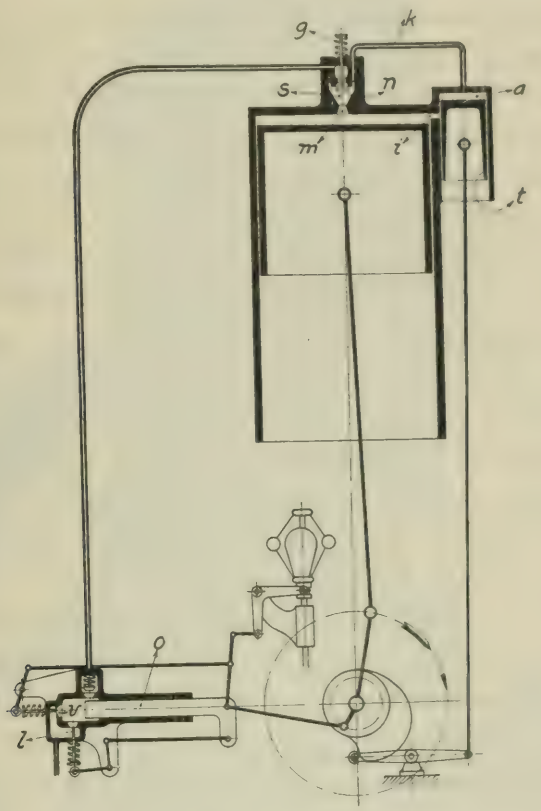


FIG. 27 SETZ OIL ENGINE WITH AUXILIARY INJECTION AIR PISTON

when the first attempts were made to burn heavy coal tar oils in the Diesel engine. Only after the engine had been in operation for some time, and was therefore warmed up, could the injection air pressure be raised sufficiently to get somewhat near complete combustion, but even then frequent "misses" or late explosions occurred. This difficulty could be successfully overcome only by injecting first a very small charge of a lighter oil, immediately followed by the charge of coal tar oil. Practically all the leading Diesel engine manufacturers in Germany have recently adopted this expedient, and quite a number of

ingenious constructions have within the last year or two been developed for this purpose. In all these schemes the fundamental idea is to inject, with the very first particles of injection air, oil particles which will immediately be vaporized and ignited, before enough of the cold injection air enters the combustion chamber to cool the air there materially. Even for ordinary fuel oils this is a prime requisite in the design of atomizers and injection nozzles to counteract, without resorting to excessively high compression, the cooling effect of the injection air, especially at light loads.

47 It is evident that this phenomenon will at once be eliminated if the injection air is obtained as illustrated by the Trinkler and Haselwander engines, or by any other process which will produce heated injection air; thus in the Trinkler engine, with the compression pressure considerably lower than in the Diesel engine, self-ignition was successfully obtained from the start. Heated injection air, however, introduces the complications dealt with in discussing the method of depositing fuel in front of the open injection nozzle at the beginning of the compression stroke, or for that matter, in any engine where fuel (or fuel vapors) and air come into contact with each other before the actual injection period. This method of introducing the oil will, therefore, have to be abandoned in favor of a scheme for keeping it entirely separate from all air until the injection period begins.

48 There is another consideration which speaks for such a step. It will be noticed that the injection process of all these various types of engines distinctly resolves itself into the two phases alluded to in Par. 16. As a result the engine makes at least one-half a revolution between the time where the measured quantity of oil is delivered and where its energy is liberated in the engine cylinder; in other words, the development of the indicator diagram is not positively controlled by the governor. In order to accomplish this the oil admission period and the injection period must fall together; regulation will then be similar to that of the steam engine, where the energy carrier is admitted and controlled by the governor at the beginning of the working stroke.

49 Under this condition it should then be possible to use injection air of a higher temperature than is now feasible, and as a result the compression pressure in the cylinder may be materially lowered. It is true that this will slightly lower the thermal efficiency; on the other hand, it must be remembered that, at least for small units, extreme efficiency is not so much the desideratum, especially in this country, as extreme simplicity and reliability of operation, which are, of course, very dependent on the working pressures.

50 It seems probable that the introduction of the fuel into a blast of injection air at the moment fuel injection is to take place, instead of employing the old method where the air blast has to pick up the fuel, would, aside from the advantages just mentioned, make it possible to use injection air of a lower pressure. This induced the author

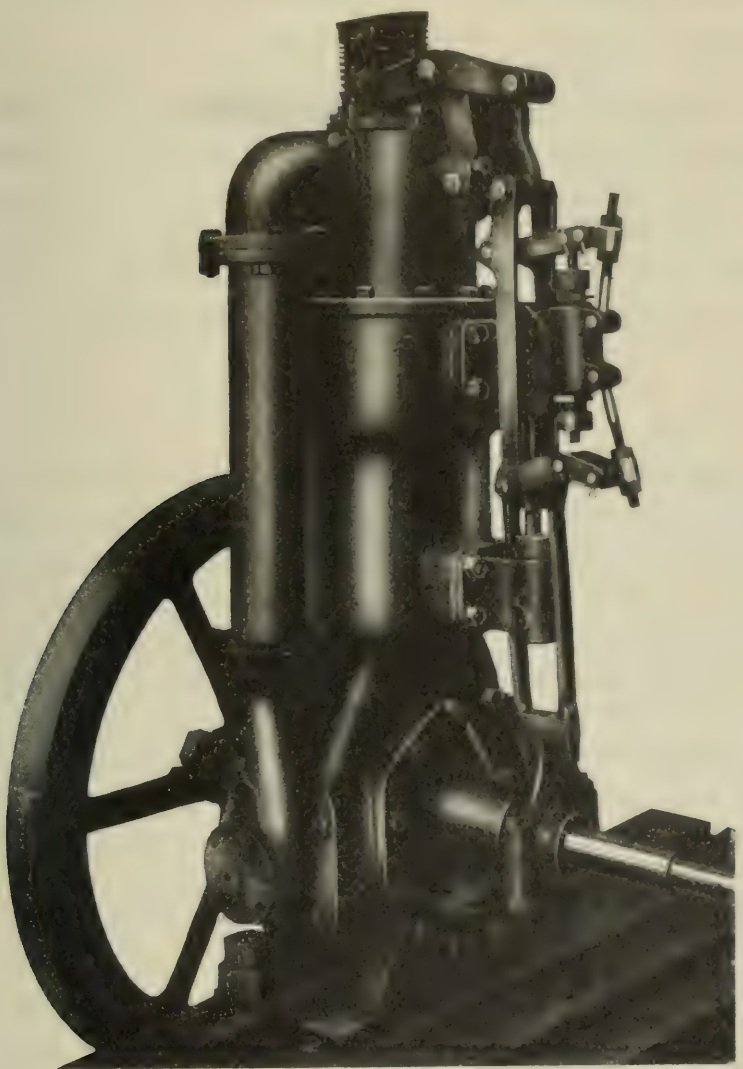


FIG. 28 SETZ ENGINE STANDING ON TESTING BLOCK

some two years ago to build a simple apparatus by means of which either one of these two methods could be used. Since the observations to be made were of a relative nature only, the air and oil spray was merely blown into the atmosphere, although a series of such experiments conducted under actual working pressures, with various shapes of nozzles, would undoubtedly help to throw much light on a

subject on which, judging from the many theories prevalent among designers of modern oil engines, little actual knowledge seems to exist. Unfortunately the limited time and means available did not permit the author to pursue his experiments as far as at first contemplated, nor to find a way to determine positively the important although elusive degree of atomization. After several futile attempts this was finally judged merely by two independent observers. Under these conditions few data of actual value could be obtained; the fact, however, has been positively established that in the case of the gradual introduction of the fuel into the injection air blast, considerably less energy was required to atomize it completely than under the old method. This was especially apparent when the oil was introduced in the form of a very fine film, thus affording a large surface over which the "disintegrating" of the oil could take place.

SETZ ENGINE

51 On the strength of these observations, as well as on the foregoing critical analysis, an experimental engine was subsequently built, the general scheme of which is shown in Fig. 27, representing the injection period. The principle of obtaining injection air is the same as that described in Pars. 32 and 33 (Figs. 19 and 20). The auxiliary piston *t* has just produced injection air of sufficiently high pressure to obtain the desired velocity through passage *k*, from whence it is directed into space *s* in such a way as to cause it to circle around its wall down towards nozzle *m*. At this moment pump *o* begins gradually to force the required quantity of oil into space *s* at a velocity determined by the tension on spring *g* of valve *n*. The oil is therefore forced directly into the stream of injection air, the velocity of which is relatively much higher than that of the oil, and the resulting abrupt acceleration of particle for particle of oil produces a complete spray at a minimum expenditure of energy. An important feature of this arrangement is the fact that valve *n* admits the fuel in the form of a very fine, cone-shaped film, thus distributing it equally over the whole surface of the injection air blast. This introduction of the fuel continues until oil pump valve *v*, which is under control of the governor, opens, when valve *n* will automatically return to its seat and close off the oil passage, or rather the oil contained therein, from all contact with air until the next injection period begins. The functions of valve *n* are thus three-fold:

a To determine the velocity with which the oil is to enter space *s*.

b To distribute equally the oil introduced over the whole surface of the injection air stream.

c To prevent the possibility of air and oil coming in contact with each other except during the injection period.

52 It is evident that in this engine the duration of the injection period is determined by the oil pump *o*; provisions must therefore be made to start the admission of fuel under all loads at the same point, relatively, to the position of the main piston. This is accomplished by means of valve *l*, which always closes at a fixed point of the pump stroke when the delivery of oil begins and continues at a rate determined by the diameter and stroke of the plunger.

53 Fig. 28 gives a view of this engine standing on the testing block. In the near future the writer expects to be in a position to give a detailed description of the constructive elements used, and to produce the data obtained by the experimental investigations now going on. What little has already been obtained promises to prove the practicability of the scheme depicted by Fig. 27, which, in its practical form, represents an engine of surprising simplicity.

54 In conclusion the writer wishes to emphasize the fact that the design and manufacture of oil engines is preëminently a matter of detail work, which might appear insignificant compared with the general problems involved in the design of an engine. However, the history of the development of the Diesel engine shows that it is just these details requiring, if properly conceived, no small amount of engineering skill in their construction, that determine the practicability of the modern oil engine; aside from the purely analytical or speculative problems involved in these details, the selection of proper materials and "fits," which experience and close observation alone can teach, are matters of utmost importance, and only the very best of tools and extremely accurate workmanship, together with broad-minded business principles, will make it possible to reach the high standard necessary to attain success.

55 Looking over the situation of the oil-engine industry in this country, where conditions for extended activity in this field are probably more favorable than elsewhere, it must be admitted that very little has so far been accomplished. Our patent records are not wanting in evidences of interest, nor have manufacturers overlooked this prime mover. Of far greater moment than the patent specification

is a full appreciation of the importance of details, the development of which along sane principles must be the designer's foremost aim. That this may involve a considerable amount of experimental work should form no barrier, for the commercial possibilities are well worth every earnest effort. There is a growing demand in this country for reliable engines that will burn efficiently the heavier hydro-

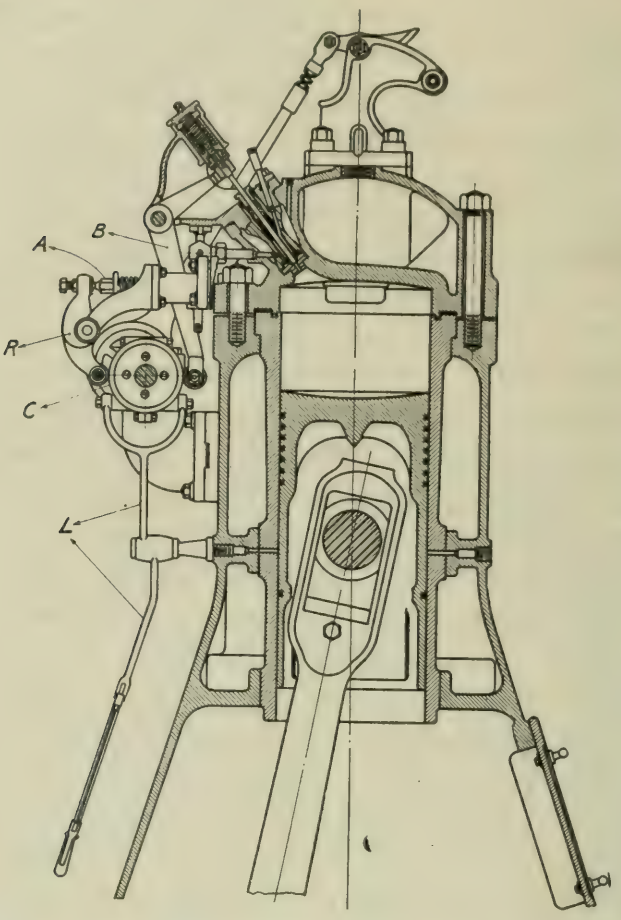


FIG. 29 CROSS-SECTION OF ATLAS DIESEL ENGINE

carbons, and it is safe to predict that the near future will see great strides towards the advance of these prime movers.

56 During the summer of 1911 the Atlas Engine Works in Indianapolis brought out a Diesel engine involving some novel features. Fig. 29 shows a cross-section through the cylinder. All valves are located in the cylinder head, the inlet and exhaust valves side by side in a vertical position, a practice which is coming more and more in favor in larger vertical gas-engine construction also. By this arrangement a combustion chamber free of any pockets is obtained, which, of

course, is conducive to high thermal efficiencies, at the same time eliminating eccentric and uncontrollable strains on the cylinder or cylinder head. Inlet and exhaust valves are operated by means of eccentrics and wiper cams.

57 The air starting valve A is placed horizontally, on the side of the cylinder head. It is operated in the usual way by shifting a cam C, by means of lever L, thereby cutting out the injection valve gear and putting in operation the air valve lever R.

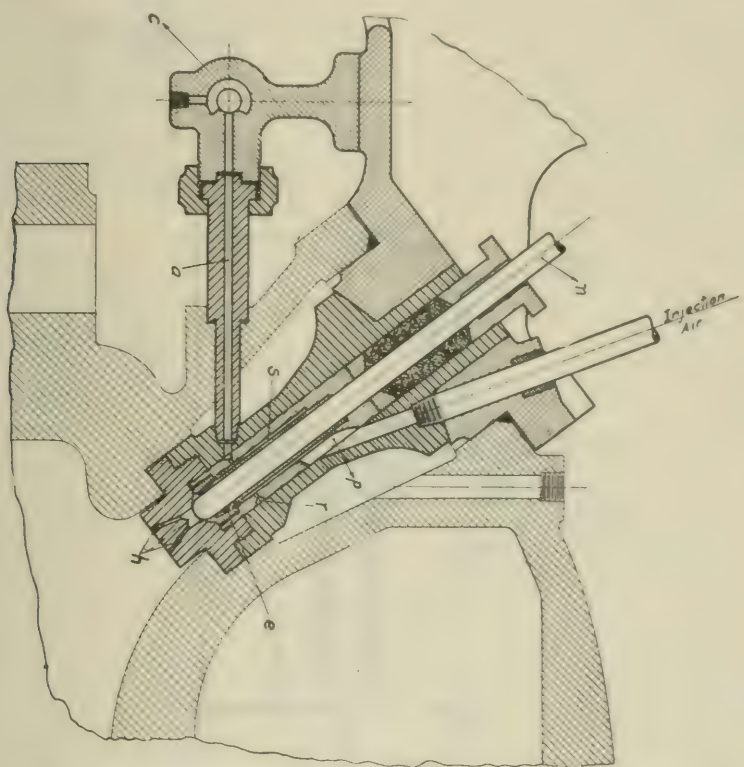


FIG. 30 INJECTION VALVE OF ATLAS DIESEL ENGINE

58 The location of the injection valve cage off center and under an angle is somewhat unusual, but is a good solution of the rather difficult problem of placing this valve as close to the center as possible without endangering the circulation of cooling water in the cylinder head. Furthermore, it makes it possible to operate this valve in the most direct way, by means of a bell crank B.

59 Fig. 30 shows a cut through the injection valve. Fuel oil enters through a into the ring space r, from where it is forced through a series of small holes into the annular space s; this same space is in communication through passage p with the injection air supply. As soon as needle valve n is lifted off its seat injection air and oil are

forced through a circle of holes *e* and through passages *h*, which here take the place of the injection nozzle, into the combustion chamber. A test cock is provided at *c*, by means of which the oil supply pipe

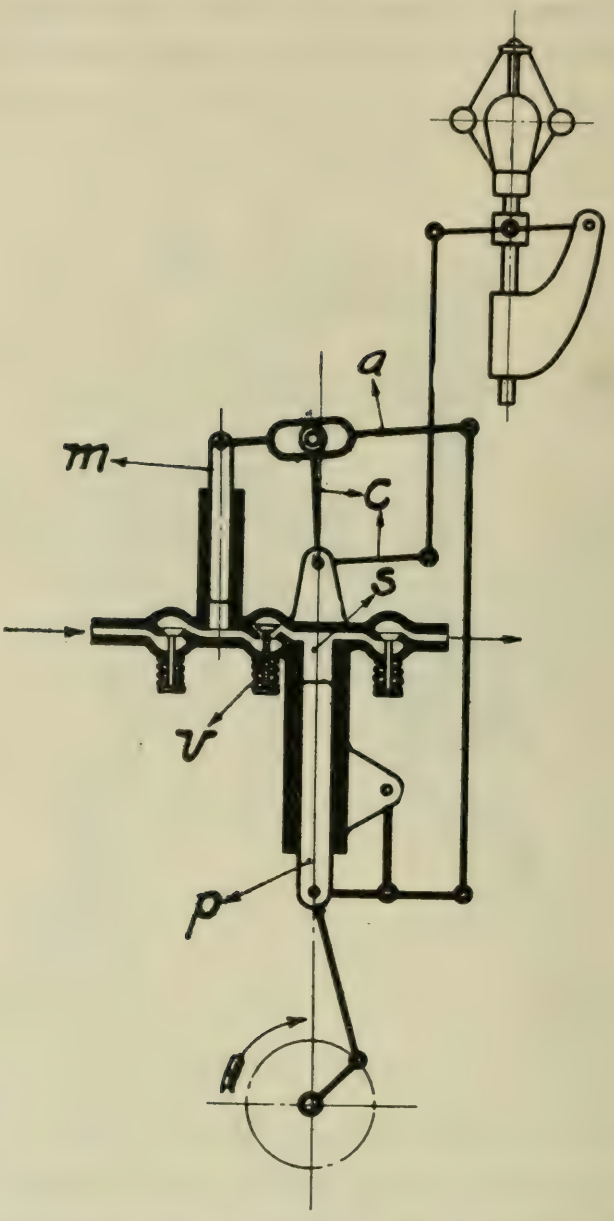


FIG. 31 DIAGRAMMATIC VIEW OF OIL PUMP OF ATLAS DIESEL ENGINE

may be filled, and the proper delivery of oil ascertained before the engine is started up. For this purpose the oil pump may be disconnected from its driving gear on the engine shaft and, after cock *c* has been opened, operated by hand until oil emerges from the cock

when the latter is closed and the oil pump connected to its driving gear again.

60 The fuel pump, a diagrammatic view of which is shown in Fig. 31, is of entirely novel construction. Aside from the usual plunger p which forces the oil into the injection valve cage, a second plunger m is used, the movement of which is derived from plunger p by a system of levers. The stroke of plunger p is constant, while that of m varies according to the position of the governor, which shifts the fulcrum of lever a by means of bell crank c or its equivalent. Plunger m is used merely to measure the desired quantity of fuel oil and to force it past valve v into space s; since the latter function is augmented by the suction of plunger p, the spring tension of valve v, which must be high enough to keep this valve closed on light loads when there will be a partial vacuum in space s, is easily overcome

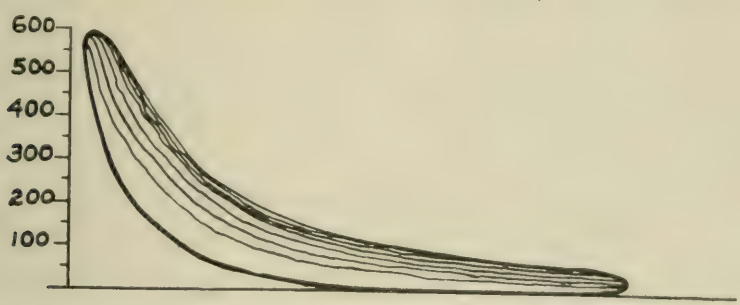


FIG. 32 DIAGRAMS OBTAINED ON OTHER DIESEL ENGINE WITH LOAD SUDDENLY THROWN OFF

practically with no pressure exerted upon plunger m except that corresponding to the slight head under which the oil enters the pump. This pressure is so slight that not even a gland is needed on plunger m, which, of course, reduces friction to a minimum. By this arrangement practically no reactions manifest themselves upon the governor, thus insuring quickness and accuracy of regulation.

61 A series of diagrams, obtained with the load suddenly thrown off, is shown in Fig. 32. In studying these diagrams it must be borne in mind that at the time they were obtained the engine was equipped with an ordinary Porter-Allen governor.

62 It remains to be seen whether the economical possibilities of the Diesel engine are such as to warrant its extensive use in the United States. The statement has often been made that, owing to the low cost of our liquid fuels, the Diesel engine cannot successfully compete with other forms of internal-combustion engines, the gain in fuel consumption not being high enough to outbalance expenditures in other

TABLE 2 REPRESENTATIVE OPERATING RESULTS

		Gas Producer Plant			Natural Gas Engine			Low-Pressure Oil Engine			Diesel Engine		
		Full Load	¾ Load	½ Load	Full Load	¾ Load	½ Load	Full Load	¾ Load	½ Load	Full Load	¾ Load	½ Load
1	Fuel per h.p-hr.	1.25 lb.	1.5 lb.	1.8 lb.	10 cu. ft.	12 cu. ft.	15 cu. ft.	1.00 lb.	1.25 lb.	1.6 lb.	0.50 lb.	0.55 lb.	0.60 lb.
2	Fuel per h.p-yr. (4500 hr.)	2.5 tons	3.0 tons	3.6 tons	45000 cu. ft.	54000 cu. ft.	65700 cu. ft.	643 gal.	803.5 gal.	1028.5 gal.	321.5 gal.	353.5 gal.	386 gal.
3	Cost of fuel	$4 per ton			30 ct. per 1000 cu. ft.			3 cents per gal.			3 cents per gal.		
4	Cost of fuel per year	$10	$12	$14.40	$13.50	$16.20	$20.25	$19.30	$24.10	$30.85	$9.65	$10.60	$11.58
5	Cost of attendance per h.p-hr.	0.40 cents			0.25 cents			0.25 cents			0.30 cents		
6	Cost of attendance per year	$18			$11.25			$11.25			$13.50		
7	Lubricating oil per h.p-hr.	0.006 pint			0.006 pint			0.006 pint			0.007 pint		
8	Cost of oil per year at $0.25 per gal.	$0.84			$0.84			$0.84			$0.98		
9	Scrubber and cooling water per h.p-hr.	8 gal.			5 gal.			5 gal.			4 gal.		
10	Cost of water per year at $0.30 per 1000 cu. ft.	$1.44			$0.90			$0.90			$0.72		
11	Operating expenses; items 4,6,8 and 10	$30.28	$32.28	$34.68	$26.49	$29.19	$33.24	$32.29	$37.09	$43.84	$24.85	$25.80	$26.78
12	Saving by Diesel engine	$5.43	$6.48	$7.90	$1.64	$3.39	$6.56	$7.44	$11.29	$17.06	.....	.....	.....
13	Interest, depreciation and maintenance respectively in per cent of investment	6+7+2=15			6+7+2=15			6+7+2=15			6+10+3=19		
14	Assuming $80 initial cost per h.p. the yearly fixed charges	$12			$12			$12			$15.20		
15	Excess cost of Diesel engine	.....			.....			.....			$3.20		

directions. To the writer's knowledge, no conclusive proof of this statement has ever been produced unless the apparent failure of the Diesel engine to secure for itself as firm a foothold in our market as in the European is to be taken as such. However, there are good reasons to believe that this condition emanates from sources for which the Diesel engine as such is in no way responsible. The following comparative investigation into the cost of power generated by an anthracite gas producer plant, a natural gas engine, a low-pressure oil engine and a Diesel engine might therefore be of interest.

63 Table 2, items 1 to 11, shows representative operating results pertaining to units of from 100 to 150 h.p., at full, $\frac{3}{4}$ and $\frac{1}{2}$ load. These figures should be fairly accurate for units considerably below and above these capacities, especially since relative values only will be used. There is no reason why results obtained in a first-class plant, under the care of a level headed attendant with ordinary experience, should under the assumed costs for fuel, labor, water and oil vary to such an extent that a fair compilation, to represent average practice, could not be made, and the figures contained in Table 2 are to be so considered. For special cases, where local requirements and conditions are known, corresponding allowances will, of course, have to be made.

64 In view of test results frequently published in trade journals and other publications some of the figures assumed for the fuel consumption per b.h.p-hr., item 1, might not find general approval. Thus, it is not uncommon to find statements to the effect that an anthracite gas producer plant will develop a b.h.p-hr. on 1 lb. of coal. There is no question but that such results are being attained on test blocks, where both engine and producer have expert attendance and run under the most favorable conditions. However, it is a well-known fact that the efficiency of a producer plant is extremely susceptible to the manner of operating. Variations in the grade of fuel, wasting of coal in charging and in removal of ashes, fluctuations of the load and above all, variations in the human element on which the maintenance of those conditions which make for the efficiency of the plant depends, all help to cause, in many cases, quite considerable variations between test results and actual operating results. This applies not only to a producer plant but also in a lesser degree to the others.

65 Wherever the proper mixture of combustibles has to be formed during the suction stroke, variations in the composition, pressure and temperature of the gas, or oil vapor in low-pressure oil engines, as well as variations in the atmospheric conditions of the air, require cor-

responding corrections in the port openings. There is no method of regulation, and never will be, which will automatically take care of all these variations in such a manner that the engine will at all loads deliver its work at the best possible economy. This must always depend to a large degree on the intelligence of the operator.

66 On the other hand, the cycle of operation renders the Diesel engine entirely free from all such influences. Variations in the atmospheric condition of the air are at once eliminated by the high compression, while the fact that the fuel is injected in liquid form directly into the combustion chamber eliminates variations of temperature or pressure. The only variable which thus remains is the composition of the fuel, which ultimately manifests itself in a smaller or larger amount of heat energy liberated in the combustion chamber and this is easily and entirely automatically handled, by the governor, provided the necessary amount of air to sustain complete combustion be present. About this there can be no question, for at all loads below the maximum there will always be an excess of air in the combustion chamber. This entire absence of uncontrollable influences in the Diesel engine accounts for the fact that in numerous installations actual operating results over extended periods are practically the same as test results; the fuel consumptions of the Diesel engine, as given in item 1, are therefore extremely conservative figures, as they represent by no means the best or even the average results on record.

67 Item 2 shows the fuel consumption per horsepower per year. The number of hours of operation per year is assumed to be 4500; for most installations this figure is, undoubtedly too high and 3000 hours would probably cover average conditions better. There are, however, plants where operation is practically continuous and although in such cases it is usually the practice to have two or more units which are run alternately, the total number of hours of operation often reaches into 4000 per year. The assumed 4500 hours should, therefore, suffice to take in practically all cases for the size units here considered.

68 In regard to the cost of fuel, item 3, it must, of course, be borne in mind that the figures given are subject to considerable variations according to localities. Anthracite producer plants are today almost entirely restricted to the eastern states, where in most localities coal in pea or No. 1 buckwheat size may be had at from $3.50 to $4.50 per ton. The assumed figure of $4, to represent average conditions, seems therefore justified.

69 Much more variation exists in the cost of natural gas. There are plants today where natural gas is available at 10,12 and 17 cents

per 1000 cu. ft. These, however, are rather rare exceptions and even in the center of the Pennsylvania gas field the prevailing figure at the present time is 28 cents, with a continual upward trend.

70 The average cost of liquid fuel was assumed to be 3 cents per gal. for both the low-pressure oil engine and the Diesel engine. This, in the writer's estimation, is a rather unfavorable assumption as far as the Diesel engine is concerned, for upon close investigation the fact could undoubtedly be established that, as a whole, Diesel engines

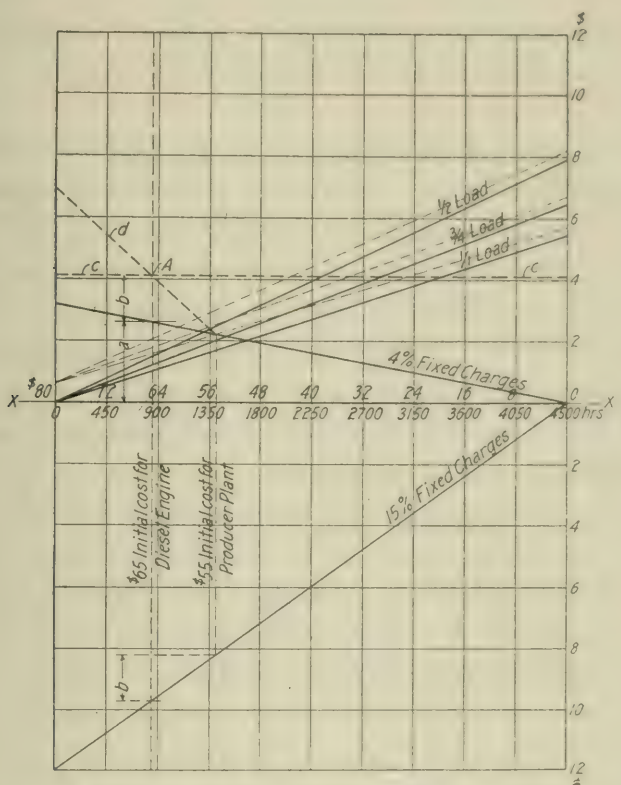


FIG. 33 COMPARISON BETWEEN THE DIESEL ENGINE AND GAS PRODUCER PLANT

are operated on a slightly lower grade and, therefore, cheaper fuel than low-pressure engines.

71 In cognizance of the generally prevailing belief, at least in this country, that Diesel engines require more skilful attendance than other forms of internal-combustion engines. the cost of labor for the former was assumed to be 20 per cent higher, item 5, which in the writer's opinion is quite a liberal allowance, for the operating requirements of a well-designed Diesel engine are today hardly more exacting than those of a gas engine, and no more repair work is needed. The latter, it is true, is likely to be of a more exacting nature than on

ordinary internal-combustion engines, but the fact must not be overlooked that the necessity of extensive or frequent repairs on Diesel engines is evidence of poor design or workmanship and will not occur in an engine designed along modern principles.

72 Owing to the higher pressures prevailing in Diesel engines the oil consumption, item 7, is slightly higher than in either gas or low-pressure oil engines.

73 On the other hand, the absence of any danger of premature ignitions will make it possible to operate the Diesel engine on slightly less cooling water than the other types, item 9. Both the water and lubricating oil consumptions are assumed to remain constant from full to $\frac{1}{2}$ load.

74 The fixed charges per year, items 13 and 14, will remain the same, no matter how many hours the plant is in operation. The assumed initial cost of the installation of $80 per h.p. is, of course, too high, even for a Diesel engine, especially since this figure is to include the engine only, and in the case of a producer plant also the gas producer, together with all piping and accessories, ready for operation. Foundations, electric generators and buildings are not to be included as these items, and with the exception of the producer plant, will vary little for the different types considered.

75 It will be noticed that the fixed charges include maintenance which, strictly speaking, ought to be charged to operating expenses, since this is a factor depending primarily upon the hours of operation of the plant. However, a study of Figs. 33, 36 and 37 will show that no appreciable error is committed by including this item in the fixed charges. Maintenance of the Diesel engine is assumed to be 50 per cent higher than for the other types. Depreciation is also assumed to be almost 50 per cent higher, 10 per cent against 7 per cent for the other. Thus, against a total of 15 per cent fixed charges in the case of the gas and low-pressure oil engines, the Diesel engine will be charged with 19 per cent.

76 The saving in operating expenses in favor of the Diesel engine, and the excess fixed charges against it are given in items 12 and 15 respectively. These are the two factors which will ultimately determine the economical limitations of the Diesel engine as against other types of internal-combustion engines. Fig. 33 shows a comparison between the Diesel engine and the gas producer plant. The hours of operation are marked in equal increments, starting with 0 on the left up to 4500 on the right, on the abscissa XX. The vertical distances from the XX axis to the full lines marked full load, $\frac{3}{4}$ load and

½ load represent savings in operating expenses in favor of the Diesel engine. For all practical cases these savings will decrease proportionally with the decreasing number of hours of operation so that these lines, obtained by connecting the points designating the savings at 4500 hours with the point 0 hours on the XX axis, will give a ready means to read off the saving for any number of hours of operation.

77 Starting from right to left the abscissa is also divided into equal parts from 0 to \$80, representing investment. The line below the abscissa marked 15 per cent fixed charges represents the increase of fixed charges from 0 to \$80; the additional 4 per cent fixed charges for the Diesel engine are plotted likewise above the abscissa. From this it will be seen that the assumed first cost of \$80 per h.p. of the installation is immaterial, as from the diagram we can easily obtain the fixed charges for any other initial cost. An example will make this clear. We will assume that the Diesel engine complete, as outlined above, costs \$65 per h.p. while the producer plant, including the producer, will cost \$55 per h.p. The location of these two initial costs is represented on the diagram by the vertical dashed lines which are marked accordingly. It will be noticed that the excess fixed charges of the Diesel engine over the producer plant consist of 4 per cent of \$65, equal to distance a, plus 15 per cent of \$65–55, equal to distance b, which, added together, will locate point A. The horizontal line CC through point A will then show at a glance where the saving in operating expenses outbalances the excess fixed charges.

78 There is, however, one point that has not yet been considered, the standby losses of the producer plant. According to the available data a fair estimate would be that the coal consumed per idle hour and h.p. is about 3 per cent of the coal consumption per h.p-hr. at full load, i.e., 3 per cent of $1\frac{1}{4}$ lb. The standby loss per h.p. per year, if the plant were never in operation, would then amount to

$$\frac{365 \times 24 \times 1.25}{2240} \times 4.0 \times 0.03 = \$0.58$$

79 If the plant were in operation for 4500 hours per year, there would remain $(365 \times 24) - 4500 = 4260$ idle hours, which would reduce the standby losses to

$$\frac{0.58}{365 \times 24} \times 4260 = \$0.28$$

80 Since there are no standby losses incidental to the operation of a Diesel engine, the above expense items will have to be charged

against the producer plant. The light dashed lines in Fig. 33 show how this affects the savings at full load, $\frac{3}{4}$ load and $\frac{1}{2}$ load.

81 From the foregoing it is evident that the vertical distances above the horizontal line *cc* to the lines designating savings in operating expenses at various loads will at once give us the actual saving per

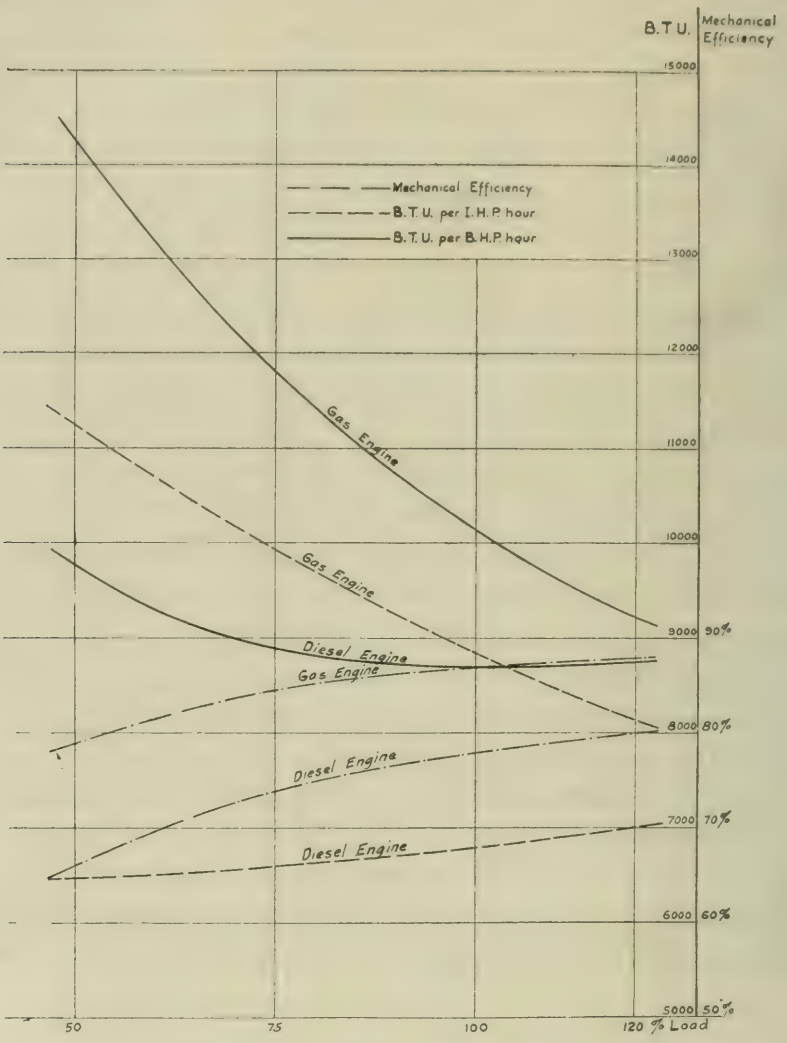


FIG. 34 MECHANICAL EFFICIENCIES AND B.T.U. CONSUMED IN GAS AND DIESEL ENGINES

h.p. per year in favor of the Diesel engine; the intersection of these lines locates the points where the saving in operating expenses and the excess fixed charges are equal, i.e., where the cost of 1 h.p. per year generated in a gas producer plant and in a Diesel engine are the same. Thus we find that at full load and considering standby losses, this point will be reached if either plant operates about 3080 hours per

year. There are, however, very few plants carrying full load right along and, except for pumping stations, the average load will in most cases hardly ever be as high as $\frac{3}{4}$ load. Fig. 33 shows that at $\frac{3}{4}$ load, and including standby losses, the Diesel engine will generate 1 h.p. per year cheaper above 2600 hours of operation than the producer plant. Assuming 3000 operating hours per year, a saving of about $0.60 in favor of the Diesel engine will be obtained for every horsepower. At $\frac{1}{2}$ load this saving would amount to about $1.60 and it would still pay to install a Diesel engine if the plant were to run only 2050 hours per year.

82 The superior economy of the Diesel engine at light loads suggested in Table 2 is made clear in Fig. 33. This is a characteristic feature of the Diesel engine and one well worth looking into a little more extensively. In Fig. 34 the mechanical efficiency, B.t.u. consumed per i.h.p. and b.t.u. per b.h.p-hr. in a Diesel engine and in the ordinary gas engine are plotted. These curves represent average values of test and operating results on 2, 3 and 4-cylinder vertical engines of from 70 to 200 h.p. published in the technical literature within the last few years. In many ways this method of averaging is of course objectionable, particularly in regard to the heat consumption, and the absolute values are by no means to be taken as representative; the idea was merely to determine the general tendency of the heat consumption for varying loads. It will be noticed that the mechanical efficiency of both the gas engine and the Diesel engine increases with increasing load, and that in the latter it is less than in the gas engine, due to the heavier pressures prevailing through a considerable part of the cycle of operation. While the b.t.u. consumed per i.h.p. increases comparatively rapidly with decreasing loads in the gas engine, that of the Diesel engine falls off. This is to be expected (see Fig. 35) since the thermal efficiency of the ideal Diesel engine is expressed by the formula.

$$\eta = 1 - \frac{1}{e^{x-1}} \times \frac{r^x - 1}{x(r-1)}$$

83 In actual practice the ratio r increases from very nearly 1 at no load to about 3 at maximum load. For

$$r = 1.5, 3$$

$$\frac{r^x - 1}{x(r-1)} = 1.09, 1.31$$

84 The mechanical efficiency divided into the b.t.u. consumed per i.h.p. gives the b.t.u. consumed per b.h.p. In both the gas and

the Diesel engine values for the latter increase with decreasing loads owing to the decreasing mechanical efficiency; but while the increase is quite rapid in the case of the gas engine the corresponding curve for the Diesel engine is very much flatter and reaches a minimum at about full load. In this respect it does not differ very much from that of a well-designed steam engine. This means that at the rated load the Diesel engine shows its maximum economy, compared with which the gas engine has no overload capacity unless it is underrated, of course, at the expense of economy.

85 This, then, together with Fig. 33, shows that the Diesel engine is preëminently suited to take care of varying loads at a much higher economy than the anthracite producer plant, and if it is remembered

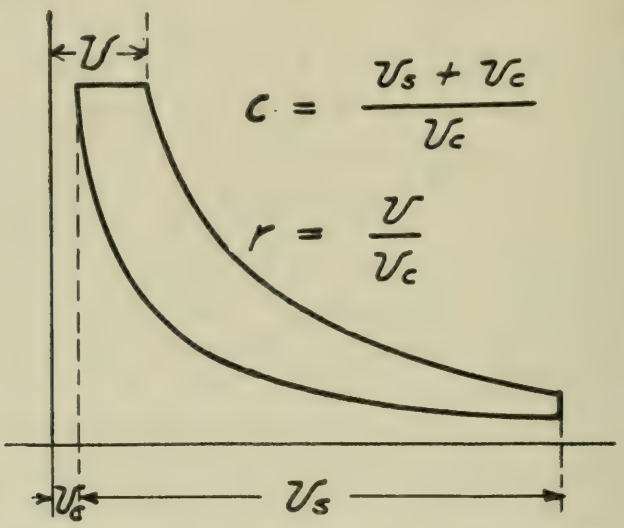


FIG. 35 IDEAL INDICATOR DIAGRAM OF DIESEL ENGINE

that in the figures underlying Fig. 33, the extra cost of larger floor space, foundation and building for the gas producer plant are not included, there can hardly be any question but that in most cases the Diesel engine deserves the preference.

86 Quite the opposite holds true in the case of a natural gas engine, Fig. 36. Assuming the initial cost of the installation to be \$40 per h.p., against \$65 for the Diesel engine, it will be noticed that with gas at 30 cents per 1000 cu. ft. things are entirely in favor of the natural gas plant. The excess fixed charges of the Diesel engine practically overbalance all its savings in operating expenses even at $\frac{1}{2}$ load.

87 Fig. 37 shows a comparison between the Diesel engine and a low-pressure oil engine. The latter designation is understood to

apply to engines working under some such principle as shown in Fig. 2, of which there is at the present time quite an extensive trade in the Middle West. The initial cost of such an engine is assumed to be $45 per h.p. against $65 for the Diesel engine. The diagram shows that at full load and 3400 hours of operation per year the cost of power produced by either unit will be alike. At $\frac{3}{4}$ load, however, this point is reached at 2250 hours of operation per year while at 3000 hours of operation the saving in favor of the Diesel engine per h.p. per year

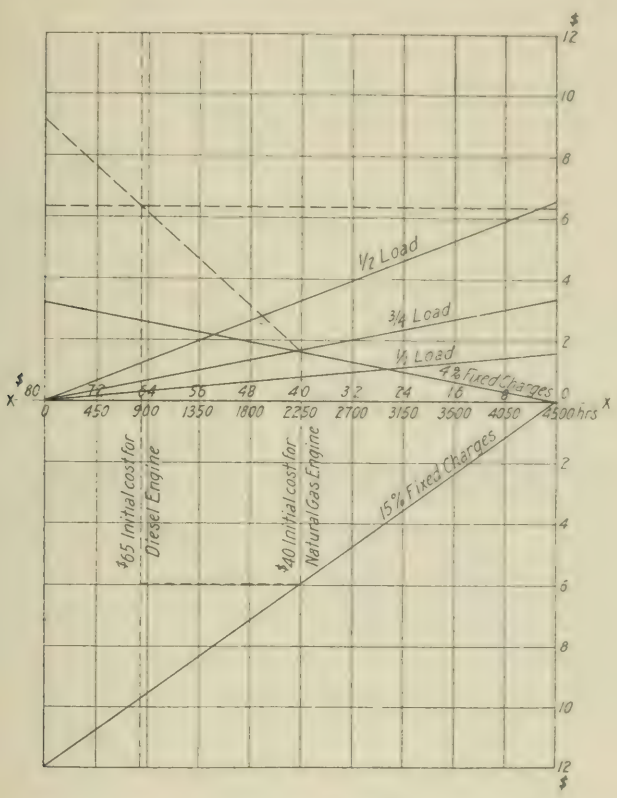


FIG. 36 COMPARISON BETWEEN DIESEL ENGINE AND NATURAL GAS ENGINE

would amount to something like $1.65, which, at $\frac{1}{2}$ load increases to $5.45 per h.p. per year. In connection with this the fact must not be lost sight of that in figuring the cost of fuel per horsepower-year for the low-pressure oil engine the cost of gasolene necessary for starting was not considered, and that the above savings in favor of the Diesel engine are, therefore, subject to an increase, depending upon the number of starts and the length of the starting periods.

88 For the sake of completeness a comparison between a Diesel engine and a soft coal producer plant should also be made. There can hardly be any question but that for larger units of say above

250 h.p. the soft coal producer will be able to hold its own against the Diesel engine; below this size, however, the comparatively very much larger cost than an anthracite producer will almost completely outbalance the saving obtained by using cheaper fuels so that the sum total of all expenses would give results hardly more favorable than those of the anthracite plant. In addition to this, the present status of the soft coal producer for smaller units is today still very

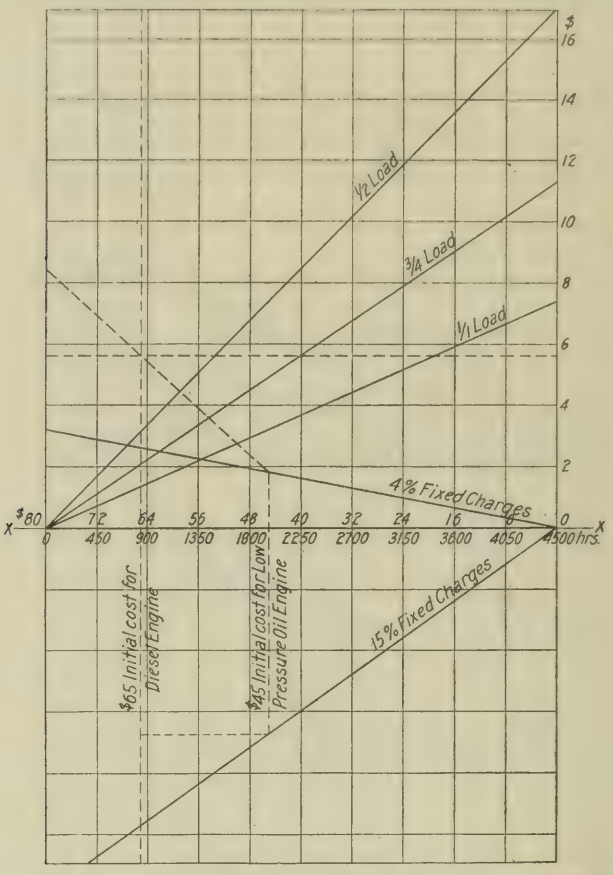


FIG. 37 COMPARISON BETWEEN DIESEL ENGINE AND LOW-PRESSURE OIL ENGINE

much in the experimental stage. These remarks apply likewise to crude-oil and peat-gas producers.

89 On the strength of the foregoing investigation it may be said that, considering average conditions, Diesel engines are in general superior to anthracite gas producer plants or low-pressure oil engines particularly for plants with varying loads and running over about 2000 hours per year. For most industrial or municipal plants where natural gas at low cost is not available, or where conditions do not warrant the installation of a gas producer plant to utilize the cheap

fuels just enumerated, the Diesel engine is therefore the most promising prime mover, even at the rather high initial cost of $65 per h.p. assumed. There is no reason why engines of this type cannot be built considerably cheaper without impairing in the least the accuracy of workmanship or resorting to objectionable features in the design. There is no better proof of this than the development of the Diesel engine during the last few years on the European continent; since the Diesel patents have expired practically every well established engine builder has taken up the manufacture of this engine, as a result of which prices have been materially reduced while design and workmanship have undergone continuous improvements.

90 Competition is a most powerful stimulant of progress, the lack of which has made itself keenly felt in the past in this particular field in the United States. A change for the better is imminent now, since a number of new concerns have taken up the manufacture of Diesel engines; if the increased activity which we are bound to witness in the near future will be instrumental in reducing the initial cost of a Diesel engine only $5 per h.p., the margin of savings in its favor will be greatly increased. A glance at Fig. 33 will readily show this. The dash line through point *A* marked *d* represents the increase of excess fixed charges against the Diesel engine from an initial cost of $55 per h.p. up to $80 per h.p. If a horizontal line, similar to *cc*, is drawn through the point on line *d* representing excess fixed charges at $60, it will at once be seen that the savings in favor of the Diesel engine increases $0.95 for every horsepower-year. The points of intersection of this new horizontal line with the line designating saving in operating expenses at full load, $\frac{3}{4}$ load and $\frac{1}{2}$ load will move farther to the left, showing that the Diesel engine becomes more economical than the gas producer plant at fewer operating hours per year than formerly where the Diesel engine was assumed to cost $65 per h.p.

Since this paper was written an admirable treatise on the same subject by Prof. A. Naegel, appeared in No. 32 of the Zeitschrift des Vereins deutscher Ingenieure, which gives a very complete review of the present status of the modern oil engine in Germany. Particularly the very latest developments of large units, as well as the provisions made for the utilization of the heavy coal tar oils, are fully described and illustrated. Although it will probably be some time before American engineers will be called upon to direct their attention towards such advanced problems, a perusal of said article is herewith recommended, especially since the many excellent illustrations are highly suggestive.

No. 1336*b*

TEST OF AN 85-H.P. OIL ENGINE

By Forrest M. Towl, New York

Member of the Society

91 A test of a De La Vergne oil engine, FH type, was made at the pumping station of the Standard Oil Company, Fawn Grove, Pa., on April 20 and 21, 1911. The engine was one of the regular 85-h.p. machines, built by the De La Vergne Machine Company, New York, cylinder 17 in. by $27\frac{1}{2}$ in. and running, as installed at Fawn Grove, at about 180 r.p.m.

92 This type of engine operates on the well-known Beau de Rochas cycle. The successive operations take place in much the same manner as in the ordinary 4-cycle gas engine, except that the fuel is injected into the cylinder at the completion of the compression stroke instead of being drawn in gradually as in the gas engine. Figs. 38 and 39, an exterior view and a longitudinal section, show the relationship of the various parts and the internal construction. Fig. 40 shows the engine and pump as installed, with clutch connection.

93 The charge of air is drawn into the cylinder through the inlet valve *A* (Fig. 39), and during the compression stroke which follows is forced into the small combustion chamber at the rear end of the cylinder, where it is compressed to about 300 lb. per sq. in.

94 A valuable feature of this engine is the high thermal efficiency without excessive cylinder pressure. The highest pressure after ignition is approximately 500 lb. per sq. in.

95 Fuel is preferably stored in an underground tank, from which it is raised by a small rotary pump driven by the engine to a miniature standpipe. An oil pump withdraws it from the standpipe and delivers it at high pressure to the spraying device, whence it is propelled into the cylinder at the proper moment in a highly atomized state.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

96 The spraying nozzle is designed especially with a view to making derangement impossible. The oil and compressed air are admitted on opposite sides of a sleeve which encloses the needle-valve pin. On the surface of the sleeve is cut a series of diagonal grooves and channels through which the oil and air are forced to pass. In this way an extremely minute subdivision of the particles of oil and a most intimate mixture with the air are obtained. The needle valve by which the charge is admitted into the cylinder is about $\frac{1}{2}$ in. in diameter, and with its appurtenances, is so arranged that the whole may be instantly withdrawn for inspection at any time.

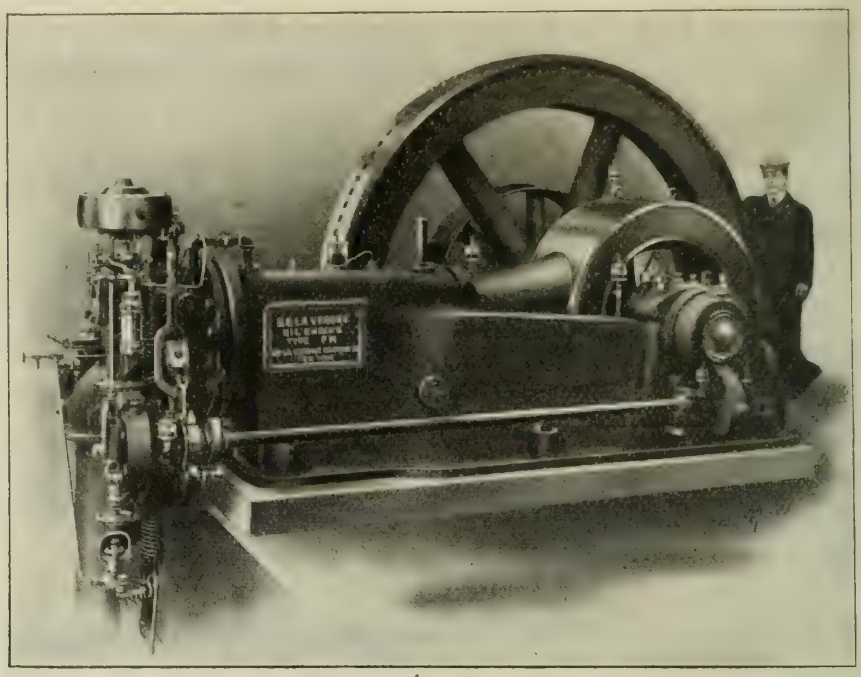


FIG. 38 EXTERIOR VIEW OF 85-H.P. DE LA VERGNE OIL ENGINE, TYPE FH

97 The air for spraying the oil is supplied by a two-stage air compressor, shown at *C*, Fig. 39, driven by an eccentric on the engine shaft. The air compressed by the first stage is stored in a tank at about 150 lb. pressure, and is utilized for starting the engine. The second stage of the compressor is quite small and handles only sufficient air to effect the spraying of the oil from stroke to stroke. The amount of air drawn in by the second stage is controlled by the engine governor to suit the various charges of fuel.

98 Ignition of the charge is effected by means of the vaporizer or hot cap shown at *D*, a device consisting of a massive gun-iron thimble,

heavily ribbed on the inside to increase its radiating surface. It is located on the side of the cylinder head and opens directly into the combustion chamber, across which and into the vaporizer the charge of fuel is injected. By this device the fuel is ignited as soon as the spraying valve is opened, and it is therefore possible exactly to time the point of ignition. As the fuel is not introduced into the cylinder until the moment of ignition, a relatively high compression may be had without the possibility of back-firing. The vaporizer must be heated by a blast lamp for a few minutes before the engine is started; but this may be removed as soon as the engine is in operation.

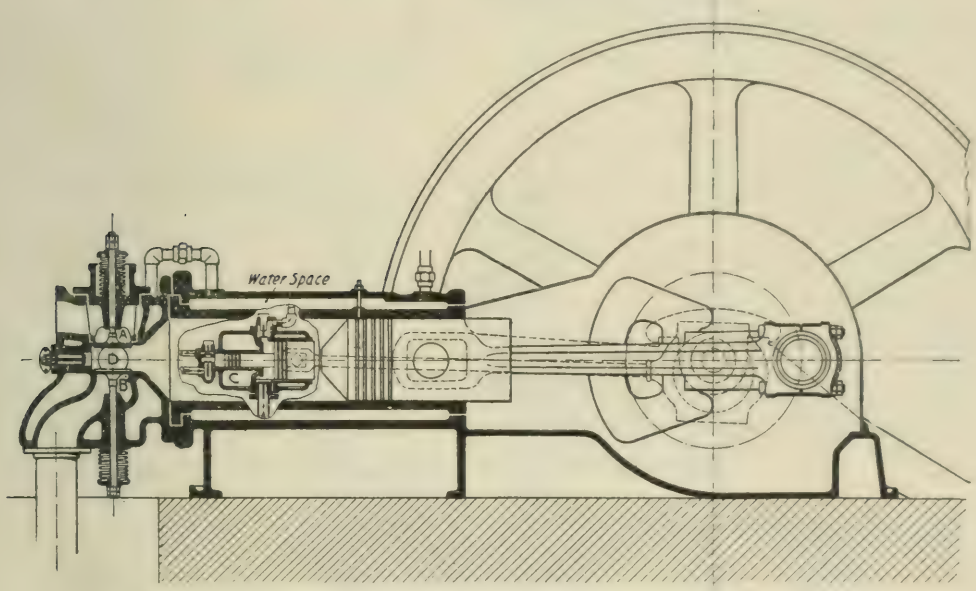


FIG. 39 LONGITUDINAL SECTION OF OIL ENGINE

99 Before shipment the engine was tested and developed a brake horsepower with 0.474 lb. of Solar fuel oil per hour when running at 65.11 b.h.p., and 0.462 lb. when running at 85.74 b.h.p.

100 In order to obtain as accurate data as possible, not only of the engine but of the combined pumping plant, it was decided to make a second brake test at Fawn Grove with the engine doing practically the same work as when pumping, and to ascertain as accurately as possible the ratio between the b.h.p. and the pump h.p.

101 In preparation for the test, a Government sealed platform scale, weighing to single ounces, was procured for weighing the oil. The water for cooling purposes was taken by gravity from a tank and

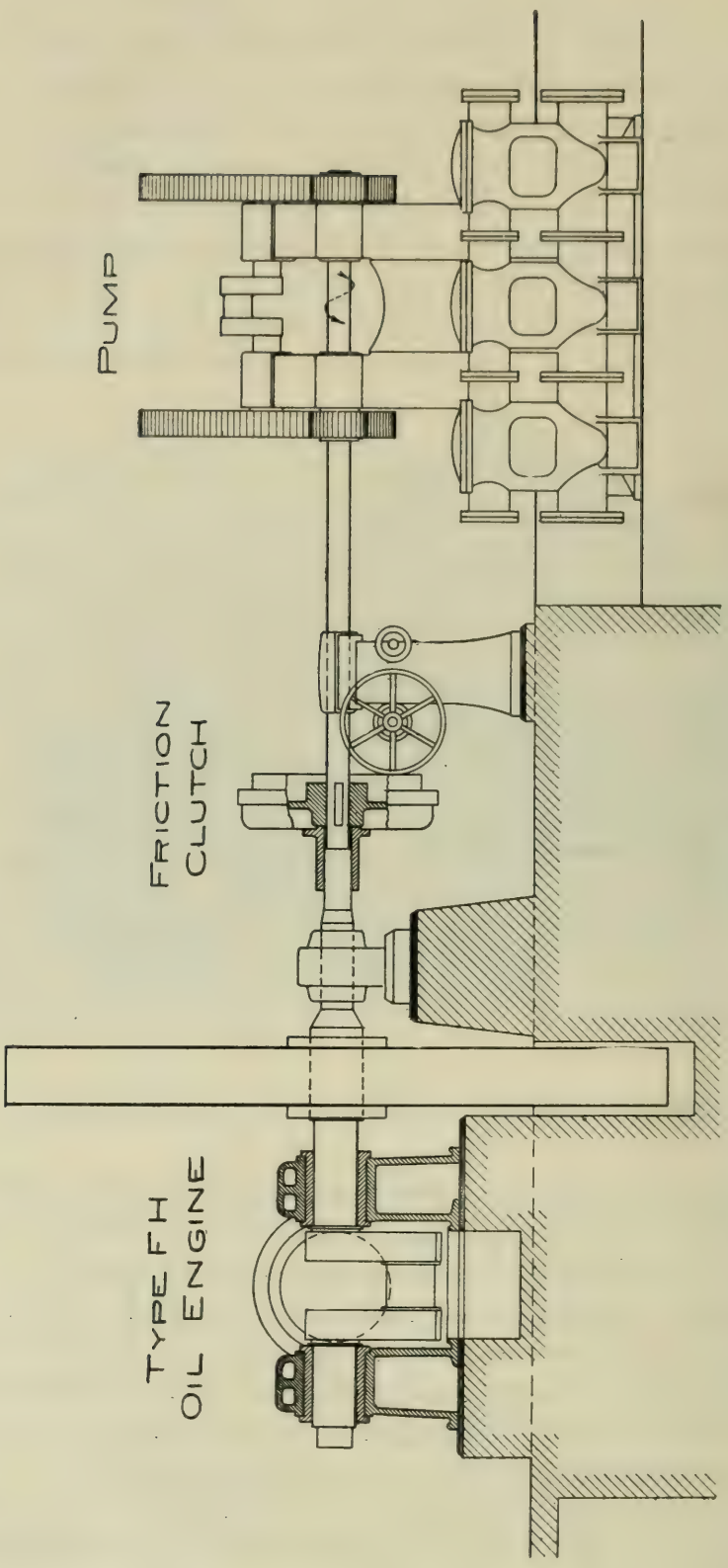


FIG. 40 ENGINE AND PUMP INSTALLED WITH CLUTCH CONNECTION

allowed to waste, the amount used being computed from measurements taken. The inlet temperature was taken at the tank, and the temperature of the water after passing the jackets by placing a thermometer in the line near the engine.

102 The amount pumped was ascertained by gaging the tank at Fawn Grove, and checked by gaging the tank into which the oil was pumped. The pressure was recorded by a Bristol recording gage and also read on a special Ashcroft gage, the latter, on the completion of the test, being taken to New York and compared with the standard gage of the company, which is graduated from a mercury column, situated in the Standard Oil Building, high enough to give direct readings up to 875 lb. per sq. in. The temperatures were taken with standardized thermometers, and the cards with a Crosby indicator, which was returned to the makers at the close of the test and found to be correct.

103 The exhaust gases were tested on the ground by using an Orsat apparatus. Samples of the oil were tested for calorific power. The average as obtained by one observer was 19,059 and this figure was used in working up the tests. Two tests were made by another observer and recorded 18,920 and 19,300 B.t.u. Prof. H. C. Sherman's formula, B.t.u. = 18,650 + 40 (Baumé deg. −10)[1] makes this 19,570. This formula is roughly applicable to all the American crude oils.

104 No analysis of the oil was made, but for the purposes of chemical calculations, it was assumed to be as follows:

	Per cent by weight
Carbon	86
Hydrogen	12
Other material	2
	100

The accuracy of the method used in analysing the gases is not such as to warrant going to the trouble of making an analysis of the oil. By comparison with available analyses the above is believed to be substantially correct.

105 In order to attach the Prony brake to the engine, it was necessary to remove the drag-link coupling between the engine and the friction clutch, thus running an extra shaft-bearing during the brake test. The brake had an arm 5 ft. 4 in. long and an unbalanced weight of 48 lb.

[1]Am.Chem.Soc., vol. 30, October 1908.

106 Three tests were made, the first, *A*, a full-load brake test; the second, *B*, a pumping test using the engine under the actual operating conditions; and the third, *C*, without disturbing any of the engine adjustments, but simply substituting the brake load for the pump load, so that the oil consumption and speed were, as nearly as possible, the same. By comparing *B* and *C* it was thought that the friction of the pump could be more accurately ascertained than in any other way.

107 The duration of each test was 3 hours, and each hour checked so closely that it was considered unnecessary to continue the runs for a longer period.

108 The air for spraying the oil was pumped by an attached compressor. There was no auxiliary machinery used, the cooling water being delivered by gravity.

109 The number of revolutions per hour was obtained by using an Ashcroft counter. During test *B* the counter was on the pump and the revolutions were computed in the ratio of the gearing; during tests *A* and *C* the counter was connected direct to the engine. The resistance of the pump load, test *B*, was so constant and the regulation of the engine so good, that the number of counts recorded for each hour was the same. The fuel consumed for the first hour was 31 lb. 2 oz., the second, 31 lb. 3 oz., and the third, 31 lb. During the brake test *C* the number of revolutions per hour recorded was respectively 10918, 10916 and 10919. The fuel consumption for the above hours was 31 lb. 6 oz., 31 lb. 8 oz., and 31 lb. 4 oz.

110 The following chemical computation was made in connection with test *C*, and is based on the analysis previously given, assuming that all of the oil was burned.

	Lb. per Hr.	
Oxygen for hydrogen combustion	30.12	
Oxygen for carbon	71.95	
Total oxygen	102.07	
Air used for combustion		443.8
Excess air (165.2 per cent)		733.2
Hydrogen burned		3.765
Carbon burned		26.983
		1207.748

111 For comparison with other pump tests the duty per 1,000,000 b.t.u. is given. This duty is, however, based on the heat units in the oil and not on the heat units delivered to the engine in the steam, as is the customary duty of a steam pumping engine.

112 It may be interesting to compare this duty with that obtained by Professor Denton in his test of the Laketon pumping engine,[1] as oil fuel was used during that test. The fuel used at Laketon contained, by Professor Sherman's formula, 19,770 B.t.u. The evaporation, test 5, was 16.64 lb. from and at 212 deg. This makes the boiler efficiency 81.3 per cent. The engine performance was 124,375, 834 ft-lb. per 1,000,000 b.t.u., or 15.985 per cent, and the total

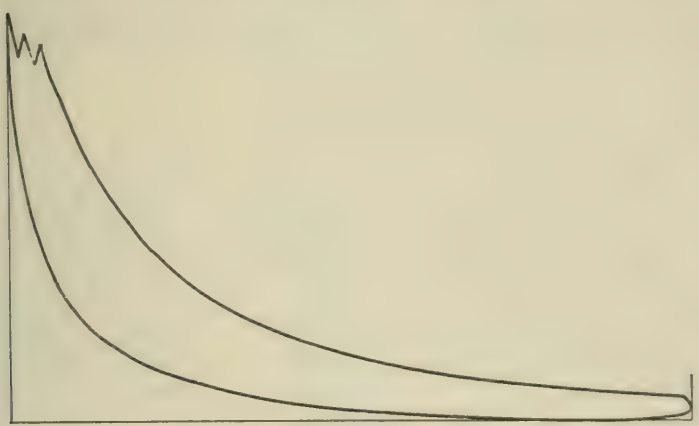


FIG. 41 TYPICAL INDICATOR DIAGRAM, TEST *A*
180 r.p.m.; 89 m.e.p.; 126 i.h.p.

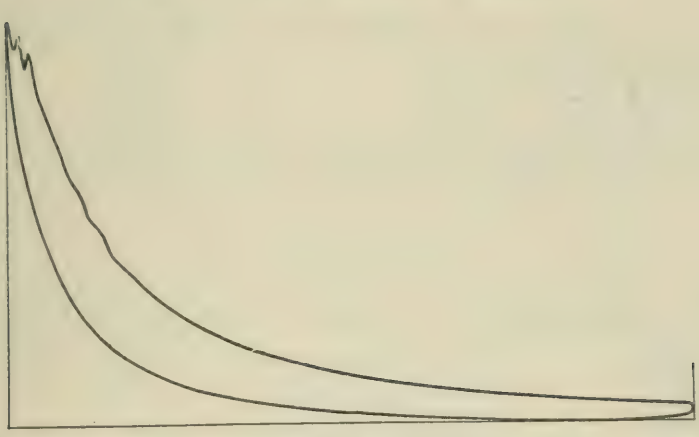


FIG. 42 TYPICAL INDICATOR DIAGRAM, TESTS *B* AND *C*
182 r.p.m.; 68 m.e.p.; 98 i.h.p.

efficiency of the plant was 13 per cent, as against 27.75 per cent for the engine and 25.52 per cent for the plant obtained from the present tests. Professor Carpenter's test of the North Point, E. P. Allis & Company pumping engine, if figured at the same boiler efficiency, would give a plant efficiency of 14.38 per cent. The Standard Oil

[1] Trans.Am.Soc.M.E., vol. 14, pp. 1349 and 1373.

Company of Louisiana have 12 of this same type of engine. Tests of two, made at Flora, La., by James Anderson, Jr., show that the engines developed a pump horsepower with 0.545 and 0.5205 lb. of crude oil per hour respectively.

113 After the engine at Fawn Grove was tested, several adjustments were made and resulted in the development of a pump horsepower with 0.48 lb. of oil per hour.

114 Fig. 41 shows a typical indicator diagram for test *A*; Fig. 42, a diagram for tests *B* and *C*. The details of the test are as follows:

HORSEPOWERS

Test number	*A*	*B*	*C*
Date	4/20/11	4/20/11	4/21/11
Start	9.00 a.m.	2.00 p.m.	9.00 a.m.
End	12.00 m.	5.00 p.m.	12.00 m.
Duration, hr	3	3	3
Average r.p.m	181.528	182.5	181.96
Average m.e.p. of cards, lb. per sq.in.	86.14	65.6	65.85
Average i.h.p.*	123.14	94.2	93.36
Prony brake load, lb	465.00		350.00
Scale load, lb	560.00		445.00
Average b.h.p	85.86	64.57 (computed)	64.68
Pressure pumped against lb. per sq. in		570.00	
Gage bbl. pumped 42 gal. per bbl		769.14	
Average gage bbl. per hr		256.38	
Pump h.p. by piston displacement		60.143	
Pump h.p. by actual gage bbl. pumped		59.48	

EXHAUST

Test number	*A*	*B*	*C*
Temperature of gases, deg. fahr	678	483	485
Average Analyses			
CO_2, per cent	7.77	5.37	5.44
CO, per cent	0	0	0
O, per cent	10.17	13.5	13.24
N, per cent	82.06	81.13	81.32
Specific heat	0.2393	0.2388	0.2387
Amount of gases			
By calculation of displacement, 70 deg. fahr. lb. per hr	1455.00	1500.00	1491.00
If temperature were same as jacket water	1181.00		1220.00
From chemical test (Par. 110)			1208.00

JACKET WATER

(Capacity of tank 39.429 gal. per in. depth)

Test number	*A*	*C*
Inches used from tank	18	$12\frac{7}{16}$
Total gal. used	709.7	490.4
Average lb. per hr.	1971.0	1362.0
Average lb. per. b.h.p-hr.	22.97	21.05
Average inlet temperature, deg. fahr.	68.7	71.3
Average outlet temperature, deg. fahr.	193.3	187.8

HEAT BALANCES

Test number	*A*	*B*	*C*
Input, B.t.u-hr.	815,153	592,813	597,976
Engine useful work			
B.t.u-hr.	218,514	164,331	164,610
Per cent	26.8	27.75	27.52
Loss in cooling water			
B.t.u-hr.	246,375		158,673
Per cent	30.2		26.5
Loss in exhaust			
B.t.u-hr.			119,520
Per cent			20.03
Loss in friction and radiation by difference			
B.t.u-hr.			155,173
Per cent			25.95
B.t.u. per hr. in cylinder work	313,391	239,739	240,046
B.t.u. per hr. in useful pump-work output of station		151,376	
B.t.u. per hr., input per b.h.p.	9,491	9,186 (calculated)	9,244
B.t.u. per hr. input per pump h.p.		9,987	
Duty, ft-lb. per 1,000,000 B.t.u. (based on oil pumped per actual gage)		198,664,000	

EFFICIENCIES

Test number	*A*	*B*	*C*
Engine efficiency			
Thermal, per cent	38.4	40.45	40.2
Mechanical, per cent*	69.71	68.55 (assumed same as test *C*)	68.55
Total, per cent	26.8	27.75	27.52
Pump			
Volumetric, per cent		98.9	
Pump and transmission, per cent		92.1	

*I.h.p. probably high due to the momentum of indicator parts and drag of pencil. This accounts for comparatively low mechanical efficiency shown in a table which follows.

FUEL CONSUMPTION

Test number	*A*	*B*	*C*
Lb. of fuel per hr	42.77	31.1041	31.375
Lb. of fuel i.h.p. per hr	0.347	0.331	0.333
Lb. of fuel b.h.p. per hr	0.498	0.482	0.485
Lb. of fuel pump h.p. by displacement		0.5171	
Lb. of fuel pump h.p. per hr. by gage bbl		0.524	

ENGINE DATA

Governor lift, in	$\frac{3}{64}$	$\frac{5}{32}$	$\frac{5}{32}$
Air pressure in tanks, lb., per sq. in	94	96	97
Regulation full load to no load, r.p.m		182–186	
Compression, lb. per sq.in		347	

LUBRICATION†

Test number	*A*	*B*	*C*
Cylinder oil			
Lb. per hr		0.6875	0.9375
Lb. per 100 b.h.p. per hr		1.0625	1.445
Engine oil			
Lb. per hr		1.78	1.16
Lb. per hr. 100 b.h.p. per hr		2.76	1.795

FUEL OIL CHARACTERISTICS

Illinois Crude Oil

Baumé	33 deg.	specific gravity	0.863
Flash point	35 deg.fahr.	burning point	65 deg. fahr.
Heating value	19,059 B.t.u. per lb. per test		
	19,570 per Professor Sherman's formula		

† A copious supply of lubricating oil was used as the engine was new. Much of this oil could have been filtered and used over again. One-third of the amount used would be the approximate need under regular operating conditions after the engine had been run for some time.

No. 1336c

DISCUSSION ON OIL ENGINES

C. E. Sargent. The paper by Mr. Setz is of special interest because it comes at a time when builders of prime movers are on the alert for something more economical than the small reciprocating engine whether steam or gas. For very large units the steam turbine has evidently come to stay, but the oil engine like the gas engine is nearly as economical in a small as in a large unit, and for this reason the power user will welcome its advent.

The constant-pressure oil engine (the term constant compression is preferable), is superior to the gas engine in having a longer range of economical load. In all my tests of internal-combustion engines I have never found as flat consumption curves as those shown by constant-pressure oil engines, two examples of which are shown in Figs. 43 and 44.

While the thermal efficiency of oil engines of the Diesel type is but a little higher than that of our most economical gas engines, yet on account of the rising price of coal, the producer losses and the over-production of fuel oil from which the gasolene has been taken, the future of the constant-pressure oil engine from a commercial standpoint is assured.

High compression can be handled as successfully as low pressure if the engine is properly designed, and on account of the elimination of the iginition system, the possibilities of premature ignition, backfiring and improper timing, the fuel oil engine working on the Diesel cycle is simpler in many respects than the gas engine.

While the methods enumerated and suggested by the author for atomizing the fuel without an auxiliary compressing outfit seem a step in the right direction and may prove perfectly satisfactory when fully developed, I understand that the largest manufacturers tried to operate in this way with the assistance of Dr. Diesel, but finally abandoned this method and went back to air storage and an auxiliary compressor.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

With this system there is the difficulty of varying the quantity of oil by the governor which is delivered against a pressure of 75 to 80

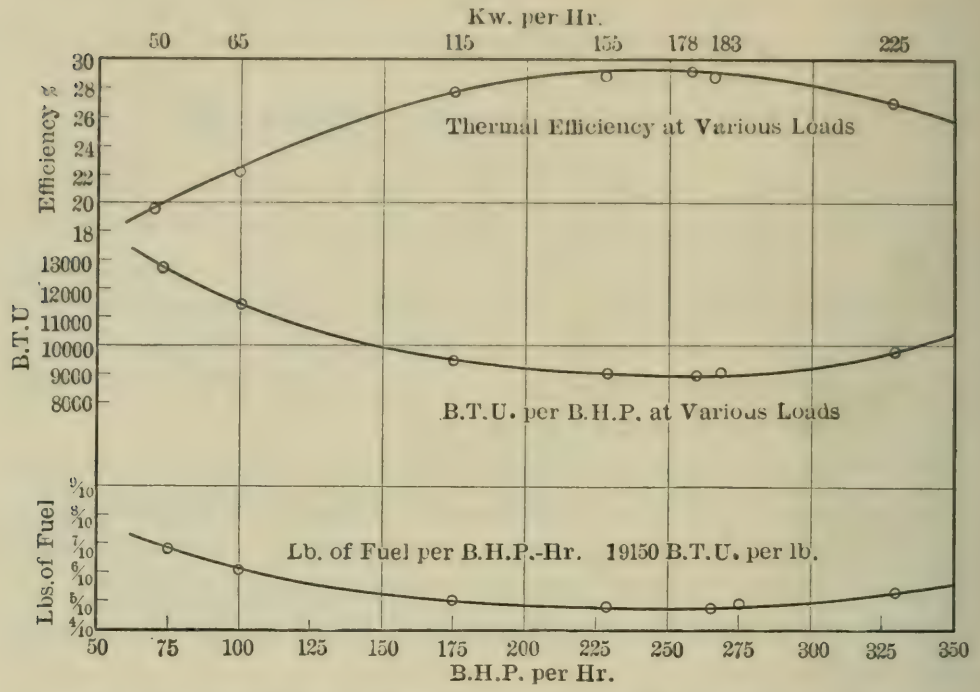


FIG. 43 EFFICIENCY AND FUEL CONSUMPTION CURVES, ATLAS OIL ENGINE, 300 B.H.P.

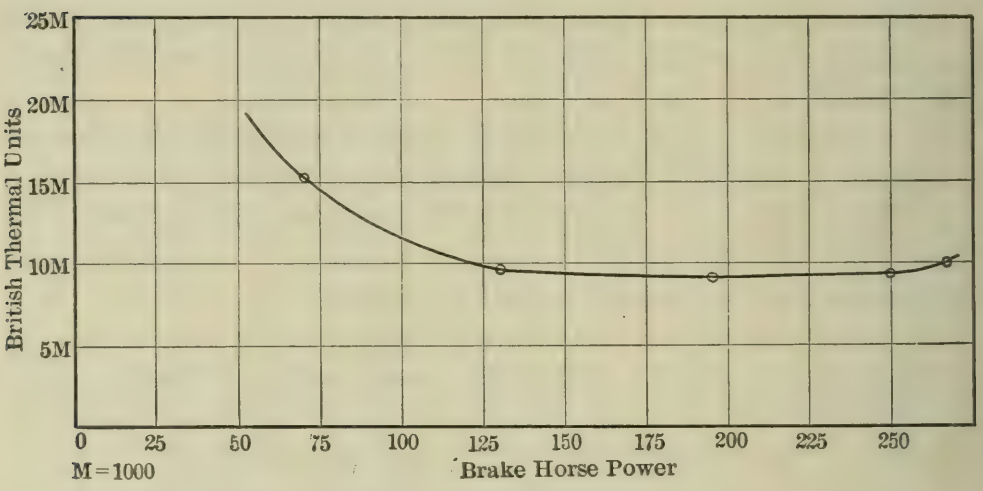


FIG. 44 CONSUMPTION B.H.P. CURVES

atmospheres, a hard mechanical problem, but solved and well protected by the leading builders of engines of the Diesel type in the United States.

If the methods described by the author of this paper are an ultimate success, then the future constant-pressure engine may be a self-contained unit, and the independent compressor, which must always run no matter how many engines in an oil-engine plant are in operation, will be used only for starting air.

A. J. Frith. To obviate the expense of the Diesel type of oil engine, the principal object according to Mr. Setz is to avoid the use of an outside compressor. Whether in doing this any trouble arises from poor combustion of the oil is not known, but it is an interesting subject and one upon which everyone wishes more information. For those who have had experience with the Diesel type of oil engine the belief is natural that the spraying of the oil by a higher outside pressure is necessary to obtain perfect results, and especially to clean the surfaces of the oil valve of oil soot. It has been reported that where liquid fuel is mechanically sprayed without this excess of air carbon is apt to deposit on the valve surfaces. Mr. Setz refers to some troubles of this kind in the types of engines he describes, but he did not describe when and where these troubles arise. None of us are too well posted on the subject, and we would like to hear in detail how Mr. Setz has overcome the difficulty of delivering mechanically liquid fuel at a high temperature through a valve without air pressure.

His problem does not seem to be one that cannot be solved and an account of the difficulties inherent in this deposit of carbon and how it has been obviated will be interesting.

In cards of some of these engines there is a rise of pressure above that of compression, especially in Fig. 25, where there are indications of explosions at the commencement of burning. The Diesel experience was that these results followed the introduction of liquid fuel imperfectly broken up by the spray and it was supposed to be due to a spheroidal condition of the oil as it entered the red hot atmosphere, and to be followed by explosive action. This indicates the trouble of injecting liquid fuels imperfectly sprayed. The view that Mr. Setz gives us, Fig. 28, of an engine built according to his system looks much simpler than many of the well-known variations of the Diesel type and also as if it deserved success.

Wm. Sangster. Mr. Towl refers to the low mechanical efficiency as found by his tests. Is this not due partly to the power absorbed by the air compressor which is a necessary adjunct to this type of

engine? Gas or oil engines exclusive of those requiring air compressors for fuel ignition, show 80 to 85 per cent mechanical efficiency, which would indicate that the use of an air compressor on this engine accounted for an additional loss of approximately 10 per cent.

Louis K. Doelling. The design of the new De La Vergne oil engine described by Mr. Setz was started about five years ago with the intention of building a reliable engine with low consumption and as low pressure as possible. From our experience with gas engines we knew that a low fuel consumption was only possible when the air and fuel in the combustion chamber were thoroughly mixed before they were ignited. To attain this condition the combustion chamber of the new engine was made cone-shaped, with the oil injection valve at the small end of the cone and the igniter in the form of a hot bulb or plate at the opposite end. The air is compressed in this chamber to 250 lb. and the oil is injected in finely divided form by an air blast having the shape of the combustion chamber. The atomized oil is mixed thoroughly with the charge of compressed air before particles of it can reach the hot plate, where they are ignited and inflame the whole mixture.

The combustion chamber is arranged transversely to the axis of the cylinder. That has proved to be especially advantageous when crude oil is burned. Crude oils contain up to 6 per cent solid matter in the form of a very fine powder that cannot be filtered out. In the De La Vergne engine this powder is deposited in the hot bulb and leaves the engine through the exhaust valve without entering the cylinder, where it would cause heavy wear on the piston and cylinder.

The engine is started like the gas engine by admitting compressed air of 100 to 150 lb. pressure into the cylinder and by relieving the compression at the end of the compression stroke. The stresses on the crankshaft when starting are not higher than when running at full speed where the maximum pressures are reduced by the inertia of the masses. The possibility of breaking crankshafts is thereby greatly reduced.

The fact that the engine can run with very low compression makes it possible to keep it running even with a heavily blowing piston, which is of great advantage in places where the engines could not be kept in proper condition on account of poor help.

Chas. Whiting Baker. In Par. 46 Mr. Setz says: "The drop of pressure of the injection air during the injection period varies according

to the load, from 250 to 500 lb.; this, of course, is accompanied by a temperature drop." Is it true that compressed air discharged through a contracted nozzle into a closed chamber, such as the clearance space of a cylinder, is lowered in temperature because of its expansion from a higher to a lower pressure?

Of course when compressed air does work against a moving piston, its energy is absorbed and it is exhausted at the end of the stroke at much lower temperature. Again, if a nozzle from a compressed-air reservoir discharges into the open air, the jet of air at high velocity does work on the surrounding air and cools itself. When the air escapes in a closed chamber, however, the jet works on a confined body of air in the clearance space and on the surrounding walls and this work is transformed into heat. It is not clear to me, therefore, that the passage of a jet of compressed air into a closed chamber is necessarily accompanied by a temperature drop. It is true there will be a temperature drop in the clearance space when the high-pressure air used in spraying enters that space; because this air before it passes through the spraying nozzle is several hundred degrees cooler than the air compressed in the clearance space. This alone is sufficient to account for the temperature drop referred to.

L. B. Lent. The able manner in which Mr. Setz has pointed out the difficulties incident to the manufacture and proper operation of oil engines should be of great assistance, not only to those who have recently commenced to build engines of the Diesel type, but also to the many who are now caring for oil engines.

It is to be regretted that many of these so-called troubles have, in the past, come to light in the purchasers' engine rooms instead of on the makers' test floor, for the impression created is nearly always adverse and oftentimes erroneous.

Oil engines have been developed abroad to a much greater degree of perfection than in this country and also to much larger sizes. This question of size or, more strictly speaking, horsepower, is a vital one if the oil engine is to take its place in central station work or any other work where units of large size are demanded. The De La Vergne engine, type FH, described in Mr. Towl's paper, is bound to suffer in this respect, although an admirable engine in others.

The foreign engines are being built as multiple-cylinder vertical units and it is the writer's opinion that both oil and gas engines will have to be built in this type where large powers are demanded and floor space is limited. Moreover, there seems to be no unsurmount-

able obstacle in the way of developing this type of internal-combustion engine. The vertical oil engine designed by Professor Junkers, which is soon to be installed in one or more ships of the Hamburg-American Line, is a very promising design of vertical engine. This engine operates on the so-called Diesel cycle and in type is a vertical Oechelhauser with tandem cylinders. The longitudinal stresses in this type are carried almost entirely by steel forgings and are practically balanced. The frame, therefore, is subject to such stress as is due only to the weights of the cylinders and other stationary parts. A study of the design of this engine shows many features which make it admirably adapted to construction in large powers.

The state of the art of building oil engines abroad is well illustrated by the contract recently entered into between Franco Tosi of Milan and the City of Rome, Italy. According to the contract, the Franco Tosi Company is to build and equip an electric power plant containing three 2000-h.p. and two 1000-h.p. Diesel engines and to sell current for a fixed sum equivalent to $15,600 per year for 13 years, plus $\frac{6}{10}$ of a cent per kw-hr. of current delivered. Penalties for interruption are imposed.

While the oil engines of the Diesel type and the FH De La Vergne engine both show excellent fuel economy and high thermal efficiency, it should be remembered that the cost of a horsepower-hour of work depends, for one thing, on the cost of fuel. Crude petroleum and its products of refining cost different amounts according to geographical location, as do the different classes of coal. It may easily happen that at a particular place the fuel cost, using coal in gas producers and engines, will be considerably less than would the fuel cost of oil in oil engines.

The figures given in Mr. Towl's paper and other data from Diesel engine performances, indicate that a consumption of 0.5 lb. of oil per b.h.p-hr. is a creditable performance except under test conditions, and may be fairly compared with a fuel consumption of 1 lb. of coal per b.h.p. in a gas engine and producer.

Under such assumptions, using the oil of 0.863 specific gravity, as given in Mr. Towl's paper, the equivalent prices of coal and oil when fuel costs per b.h.p. are the same are given in Table 3. The fuel cost per b.h.p. is also given.

From these figures, it is seen that when oil costs as low as 2 cents per gal., coal may cost $3.10 per ton of 2000 lb., and when oil costs 4 cents per gal., coal may cost $6.20 per ton. The fuel costs per b.h.p-hr. are in these cases 0.155 and 0.31 cents per b.h.p-hr. respectively.

A horizontal double-acting tandem gas engine can be bought for about $30 per b.h.p. in sizes above 250 h.p. and the necessary producer equipment can be purchased for $10 per h.p. The writer has known of oil-engine prices of anywhere from $45 to $70 per b.h.p., so unless the prices of oil engines decrease, the fixed charges against the gas engine and producer will be less than against the oil engine; although the fixed charges against the increased land and building area necessary for the producer plant may offset it. The relative

TABLE 3 EQUIVALENT COSTS OF OIL AND COAL FOR SAME COST PER B.H.P.

Cost of Oil, Cents per Gal.	Equivalent Cost of Coal, Dollars per 2000, Lb.	Fuel Cost per B.h.p-Hr., Cents
2.0	3.10	0.155
2.25	3.49	0.174
2.5	3.87	0.194
2.75	4.26	0.213
3.0	4.65	0.233
3.25	5.03	0.252
3.5	5.42	0.271
3.75	5.81	0.29
4.0	6.20	0.31
4.25	6.58	0.33
4.5	6.97	0.349
4.75	7.36	0.368
5.0	7.74	0.387

cost of fuel and of other items of power costs must be weighed in most every case, for there certainly are limitations to the use of the oil engine just as there are to the use of the gas engine, the steam engine and the steam turbine.

The choice of prime movers for any power plant still requires the same careful consideration by the engineer.

H. J. FREYN. A little study of the oil-engine situation which I have recently made shows that according to the best results published in the technical press the effective thermal efficiency of modern Diesel oil engines is in the neighborhood of about 32 to 33 per cent.

Since the Diesel engine owes its existence primarily, in fact almost exclusively, to its unsurpassed fuel economy, it is not surprising that its advent and development have had a singular economic significance

in Germany and France and on the European continent in general, where fuel is very expensive and the standard of manufacture high, whereas in England and more particularly in America where excellent cheap fuels abound, while skilled labor is scarce and dear, there is manifestly not the same inducement for the introduction of very costly although thermally highly economic machinery.

Thermo-economical considerations alone must be credited with the fact that the Diesel engine is an ideal marine engine, since irrespective of its cost, the quantity of fuel which can be carried along by vessels is limited.

Crude oil and "slop" distillates can be purchased in this country at 2 to 3 cents per gal. or 0.25 to 0.38 cent per lb. or more, according to locality. In France a very high duty is levied on all imported fuel oils. It amounts to 0.82 cent per lb. or 6 cents per gal., making the total cost of fuel oil, f.o.b. point of consumption, from 1.36 to 1.45 cents per lb. or from 10 cents to 10.6 cents per gal.; even tar oil, manufactured in France, will cost, f.o.b. point of consumption, from 0.682 to 0.820 cent per lb. or 5.5 to 6.5 cents per gal. in tank cars. In Germany the duty on crude oils and derivatives is 0.41 cent per lb. if used for power purposes; it is twice that amount when used for lighting purposes. Thus fuel oil (gas oil) imported from the United States, Russia or Roumania will cost from 0.85 to 1.16 cents per lb., f.o.b. place of consumption, corresponding to 6.6 to 9 cents per gal. The price of coal tar oil in Germany received in tank cars will vary from 0.43 to 0.46 cent per lb. or 3.75 to 4 cents per gal. In Russia and Roumania fuel oil is materially cheaper and its price will correspond approximately to the price in the United States.

This comparison shows that natural fuel oils in Germany and France are from 3 to 4 times more expensive than in this country, and furthermore that even the artificially manufactured tar oils cost at least 50 to 100 per cent more then natural oils in this country. This will explain without further comments why the oil engine and especially the Diesel engine has had such a wonderful development abroad.

In the current literature, numerous estimates can be found showing the difference in cost of installation of such plants of various capacities, particularly of less than 500 h.p.; in every single instance the different authorities reach the conclusion that the first cost of an oil-engine installation does not differ materially from that of steam or producer gas plants, in fact a number of them figure that the balance is slightly in favor of the Diesel engine, with present-day European Diesel engine prices.

Conditions have not always been so, but the expiration of the basic Diesel patents and the keen competition among European oil-engine manufacturers have reduced engine prices very materially in the last year or two as may be gathered from the following information:

In France a 250-h.p. Diesel engine running at 180 r.p.m. sold in 1905 for $18,500 or $74 per h.p. In 1911 the same engine could be purchased for $13,000 or $52 per h.p.; a high-speed engine of the same capacity running at 300 r.p.m. can today be bought in France for $9700 or $36 per h.p.; a 400-h.p. Diesel engine running at 180 r.p.m. sold in 1905 for $23,000 or $57.50 per h.p., whereas in 1911 the same engine could be obtained for $18,000 or $45 per h.p.; the corresponding high-speed engine costs only $14,000 or $35 per h.p. For a Sabathé marine engine of 36 h.p. weighing 5500 lb., $4600 or $100 per h.p. (65 cents per lb.) was asked in France in 1910.[1]

Under the existing conditions of fuel and labor cost in the United States, it is questionable whether the oil engine in large sizes will be a serious competitor of the steam turbine on the Pacific coast. The prospects for installing large Diesel engines will probably be much better in the East, especially after the opening of the Panama Canal, when oil will be available in the Atlantic coast states in large quantities and at a lower price than today, since the question of installing large and medium sized Diesel engines in the great power houses of the East to take care of peak loads will appeal much more than today as the same fuel oil will be available for raising steam for turbo-generators and for direct combustion in Diesel engines without any standby losses.

Broadly speaking the outlook for the oil engine of medium size in this country seems good, perhaps not so much in the immediate future as later on after more knowledge of the advantages of the oil engine is disseminated and its good features are demonstrated; but over-enthusiasm should be carefully guarded against, since as mentioned before, economic conditions in this country are materially different from those abroad, where the oil engine seems to have entered upon a triumphal career.

Edwin D. Dreyfus. To encourage the publishing of the best performances that engines have made, is I am certain, a very good suggestion. At the same time, however, it would be well to keep in view also the records ordinarily made so that we may have a fair

[1]Du Bousquet, Mem. Ing. Civ. France.

bench mark on which to base our normal operation. It is just such a matter that leads many of us to grievous mistakes in contemplating results which we afterwards fail to achieve regularly.

I consider Mr. Freyn has made some very fortunate remarks about over-enthusiasm. Personally, while I am an advocate of gas power, I believe the internal-combustion engine is limited in size for practical reasons. That any shop whether qualified or not by past experience should enter the oil engine field simply because it appears attractive is closely parallel to what the conditions of the producer gas field was sometime ago.

The Engineer[1] of London and Power in this country have sounded a note of warning. And I would like to quote the following from an article on Diesel Engine Difficulties, published in the former.

> In our recent leading article on the weak points of marine oil engines we intentionally confined ourselves to what we termed inherent and infantile weaknesses of the principle. There is another point which is of great importance at the present moment. It was brought out strongly in Mr. F. Shubler's paper on the Diesel engine read before the Institution of Mechanical Engineers at Zurich and is well worth emphasizing. The great possibilities of the Diesel engine have led to an enormous interest being taken in the subject, not only in the Press but by manufacturers, some of whom have had experience in building gas engines, some of whom have had none, so that the impression gained is that anyone who can make a steam engine, can equally well draw out and build a Diesel engine. It is just here that a note of warning may be sounded. The "fit" and materials necessary to produce a good steam engine are now matters of common knowledge, but when it comes to the construction of a Diesel engine, experience which alone gives the necessary knowledge, is very limited, and the patent specifications do not provide more than the smallest percentage of what is required to build an engine which will satisfactorily withstand pressure four times those which the steam engine builders have been accustomed to deal with. The successful builders of large powered engines today are those who have been at work in this direction for years, starting with quite small units and finding and overcoming fresh difficulties with each progressive step. These difficulties must be expected to occur equally in the case of the first engine of any power—they will be exaggerated when that power is comparatively high. It cannot be too strongly urged upon manufacturers that the progress of the adoption of the oil engine afloat, will only be advanced backwards by doubts being raised in the minds of the public, owing to the want of immediate success of applications, the inception of which has been heralded with the glare of trumpets, but based upon an entire want of appreciation of the great difficulties involved.

H. R. Setz. In regard to the drop of temperature during the injection period, it is true that if there were no resistance all the kinetic

[1]Aug. 4, 1911.

energy acquired by the injection air would be reconverted into heat in the combustion chamber. However, a considerable amount of this kinetic energy is spent in the atomizer disintegrating the fuel into minute particles, the amount varying according to the method used in introducing the fuel. The remainder of this kinetic energy serves to carry the fuel spray into the combustion space of the engine where it produces an intimate mixture of fuel and air and is eventually dissipated and reconverted into heat by eddy currents and friction. In view of the fact that the initial temperature of the injection air was already considerably below that of the air in the engine at the end of the compression stroke, when injection takes place, it can readily be seen that the fuel spray, when it enters the combustion chamber, must be quite cold, and what little kinetic energy remains for reconversion into heat would never be sufficient to re-establish the temperature existing before the cold fuel spray was introduced. For this reason utmost care must be taken to introduce with the very first particles of injection air particles of fuel which will immediately ignite and produce a sufficient temperature increase to outbalance the cooling effect of the following cold fuel spray.

A few remarks I wish to make concerning the manufacture of Diesel engines, for this, more than anything else, determines its practical success, in operation as well as first cost. The diagrams, Figs. 33, 36 and 37, show how the difference of only a few dollars in first cost can change things entirely in favor of or against the Diesel engine. The problem to make this engine one of our accepted prime movers will, therefore, primarily, have to be solved in the shop.

Not enough emphasis can be laid on extreme accuracy of workmanship; in order to build accurate machinery economically only first-class tools can be used, and this is one of the striking features of all the European shops where Diesel engines are being manufactured. The time will undoubtedly come when gas engine and possibly steam engine manufacturers will feel inclined to take up the manufacture of Diesel, or some other form of high-pressure oil engines, but judging from what I have been able to observe, comparatively few have today the equipment or form of organization which would enable them ever to become a factor in this particular line. The standard of refinement of the mechanical processes employed is too low in the shop where attention centers on quantity rather than on quality of the product. The usually inadequate equipment and the spirit prevailing in such shops are too serious a hindrance toward the strict observance of the close limits required in Diesel engine construction. During my

recent experience in a shop of this nature, work had to be done over and over again before the required accuracy was obtained, of course at a prohibitive cost. Diesel engines can become a commercial success only if manufactured under conditions where extreme accuracy can be obtained without putting an undue check on the rate of production. Unless a shop is equipped and managed to meet this requirement, disappointment is certain to follow.

FORREST M. TOWL. Referring to the remarks of Mr. Sangster as to the low mechanical efficiency of the engine as ascertained by the Fawn Grove test, the following is the result of computations made as to the power used by the air compressor:

1st stage		3.00 h.p.
2d stage		3.30 h.p.
	Total	6.30 h.p.
20 per cent estimated friction		1.26
		7.56 h.p.
I.h.p of engine	123.14	
Percentage used by compressor	6.14	

The above figures are based on a volumetric efficiency of the compressor of 100 per cent. If efficiency of compressor was figured at 80 per cent, there would only be about 5 per cent of the power used by the compressor.

If the compression work done outside the main cylinder had been done inside the cylinder, the i.h.p. would have been smaller and the proportion of the actual work done to that shown by the indicator would have been greater.

No. 1337

DESIGN CONSTANTS FOR SMALL GASOLENE ENGINES

WITH SPECIAL REFERENCE TO AUTOMOBILE WORK

By William D. Ennis, Brooklyn, N. Y.

Member of the Society

The following notations and expressions apply to a single-acting water-cooled cylinder in a traction engine using gasolene as fuel:

p = absolute pressure of gas or mixture, in lb. per sq. in.

v = volume of gas or mixture, in cu. ft.

T = absolute temperature of gas or mixture

D = displacement of piston per stroke, in cu. ft.

S = speed of piston, in ft. per min.

W = ideal work done per power stroke, in ft–lb.

d = diameter of cylinder, in in.

s = stroke of piston, in in.

r = revolutions per minute made by engine

N = revolutions per minute made by wheel of car

w = diameter of loaded car wheel, in in.

F = ideal tractive force, in lb.

A = head-end area, in sq. ft.

n = an empirical exponent, having a value between 1.29 and 1.38

V_0 = velocity in ft. per sec.; V = velocity in miles per hour

m = weight of car (loaded), in lb.

R = total resistance to propulsion, in lb.

l = a factor by which W may be multiplied to obtain actual work at the engine shaft

p_m = average continuous net effective pressure in the cylinder, lb. per sq. in.

u = ratio of rotative speeds (r.p.m.) of car wheel and piston $= \frac{N}{r}$

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers. Published in abstract only. The complete paper appeared in The Journal, December 1911.

2 The preliminary relations given below at once follow:

$$d^2s = 2200\,D, \quad sr = 6S, \quad \text{h.p. (at tires)} = \frac{RV}{375.5}$$

$$336V = Nw, \quad V_o = 1.462V \quad V_o^2 = 2.15V^2$$

3 Fig. 1 is taken as the reference indicator diagram for further analysis, the lines 23 and 41 being vertical, and the value of n being the same for expansion as for compression. We have

$$W = 144\left\{\frac{p_3v_3 - p_4v_4 - p_2v_2 + p_1v_1}{n-1}\right\} = \frac{144}{n-1}\left\{v_3(p_3-p_2) + v_1(p_1-p_4)\right\}$$

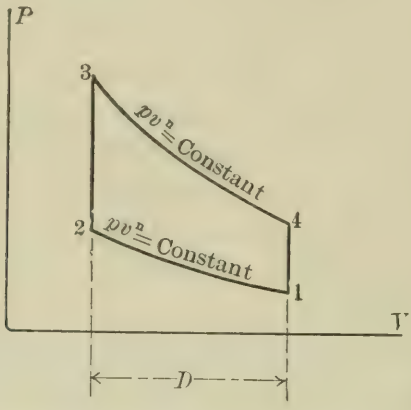


FIG. 1 REFERENCE DIAGRAM

Also, letting $\frac{v_2}{v_1} = a$

$$v_1 - v_2 = D; \quad v_2 = av_1; \quad v_1(1-a) = D; \quad v_1 = \frac{D}{1-a}; \quad v_2 = \frac{aD}{1-a}$$

$$W = \frac{144D}{(n-1)\ (1-a)}\left\{a\ (p_3 - p_2) + p_1 - p_4\right\} \quad \ldots\ldots\ldots [1]$$

4 Ignoring journal and transmission friction,

work at cylinder = work at tires

$$\frac{\pi p_m d^2}{4} \times \frac{2sr}{12} = F \times \frac{\pi wN}{12}$$

$$F = \frac{d^2 sr\, p_m}{2Nw} = \frac{1100\,Drp_m}{Nw} = \frac{3d^2Sp_m}{Nw} \quad \ldots\ldots\ldots\ldots [2]$$

and since $p_m = \dfrac{W}{4 \times 144\, D}$; we have, letting $\dfrac{W}{D} = c$

$$F = \frac{cd^2sr}{1152Nw} = \frac{1.915\, cDr}{Nw} = \frac{cd^2S}{192Nw} \quad \ldots\ldots\ldots\ldots [3]$$

5 If the car moves at a velocity V_0 as compared with that of the surrounding air, then, ideally, the resistance due to air is

$$R_a = 0.0025\, AV_0^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

6 The work done in the reference diagram of Fig. 1, per power stroke, is from equation [1]

$$cD = W \text{ ft-lb.}$$

For a single-acting 4-cycle cylinder

$$\text{h.p.} = \frac{Wr}{66000} = \frac{crD}{66000} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

Alternatively, from equation [2]

$$\text{h.p.} = \frac{FV}{375.5} = \frac{1100\, Drp_mV}{375.5\, Nw}$$

But $V = \dfrac{Nw}{336}$, and $p_m = \dfrac{c}{576}$; whence

$$\text{h.p.} = \frac{1100\, DrcNw}{576 \times 336 \times 375.5\, Nw} = \frac{crD}{66000} \quad \ldots\ldots\ldots\ldots [6]$$

$$= \frac{crd^2s}{145,200,000} = \frac{cd^2S}{24,200,000} \quad \ldots\ldots\ldots\ldots [7]$$

7 Now since:

$$\frac{p_3}{p_2} = \frac{T_3}{T_1\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}}$$

$$p_3 - p_2 = \frac{p_2T_3}{T_1\left(\dfrac{p_2}{p_1}\right)^{\frac{n-1}{n}}} - p_2$$

The rise of pressure during combustion thus varies inversely as the value of n and inversely as T_1. Since it is the condition of maximum work that both $(p_3 - p_2)$ and $(v_1 - v_2)$ should be maximum, then a low value of n is desirable. It also appears that at maximum work conditions T_1 should be a minimum, as should p_1. We have then the following concurrent values:

4-Cycle Cylinders

Type	n	p_1	T_3	T_1	p_2
Best	1.29	12	3460	600	100.0
Normal	1.32	12	3460	660	93.0
Worst	1.38	14	3460	760	45.0

Derived Values

Type	$\frac{v_2}{v_1}$	T_2	p_3	p_4	cl	(assumed)	cl
Best	0.193	967	358	43.0	12950	0.85	11008
Normal	0.212	1080	298	38.4	9786	0.80	7830
Worst	0.429	1049	148	46.2	7960	0.75	5970

These values will be used in the subsequent discussion, although an adequate series of experimental values of p_m is to be preferred.

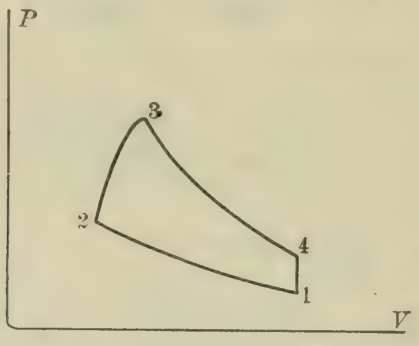


FIG. 2 DIAGRAM AT HIGH SPEED

8 If in equation [7] we substitute the values $c=9786$, $S=1000$, we obtain h.p.$=\frac{d^2}{2.49}$, a close confirmation of the $\frac{d^2}{2.5}$ formula of the Association of Licensed Automobile Manufacturers. This is horsepower gross, and the power available for propulsion will be reduced to an extent determined by the amount of cylinder loss, journal and transmission friction.

9 A rather limited group of observations indicates that the effect of journal and transmission friction may be represented by an equivalent constant loss of ideal tractive force. For a given car, nearly regardless of speed variations, the tractive force at the wheel rim is therefore taken at g lb. less than the ideal computed tractive force.

10 Since in a given cylinder the horsepower varies as fS^e

$$\frac{fS_2^e}{fS_1^e}=\frac{(R_2+g)\,V_2}{(R_1+g)\,V_1}$$

and since u varies as $\frac{V}{S}$ when the wheel diameter and stroke are fixed

$$\frac{u_1}{u_2} = \frac{R_2+g}{R_1+g}\left(\frac{S_1}{S_2}\right)^{e-1} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [8]$$

ACCELERATION CONDITIONS

11 When a car starts on a straight level road, the resistances to be considered are (*a*) friction, (*b*) head-end resistance and (*c*) that due to acceleration. In the following analysis, the first will be ignored and resistance and horsepower will be referred to the wheel rim rather than the cylinder.

12 The car starts under a constant tractive force, capable of overcoming a constant resistance, while the motor is getting up to full power and speed, the car meanwhile accelerating. As soon as maximum power is reached, further increase in velocity is possible only as the resistance decreases, acceleration becomes less rapid, and the product of the resistance and the velocity is constant. If we ignore the question of gears, or rather, if we assume a continuous variation of gear ratio to be possible, this condition of things will continue until the head-end resistance alone, at the attained velocity, consumes all the power. Acceleration will then cease and the car will thereafter move at constant speed, resistance and horsepower.

13 During the early part of acceleration, the head-end resistance is negligible. With any reasonable (wheel rim) tractive force, the fraction thereof not available for acceleration, at velocities below 5 miles per hour, may be safely ignored. Within this limit

$$\text{Force} = \text{mass} \times \text{acceleration}$$

$$\frac{dV_o}{dt} = \text{constant} = \frac{32.2\,R}{m}$$

where R is the available tractive force.

14 As an illustration, consider a touring car weighing (loaded) 3300 lb., having an equivalent head-end area of 26 sq. ft. driven by a 4-cylinder 4-cycle engine, and required to maintain a speed of 50 miles per hour in still air on a straight road, or of 20 miles per hour on a 9 per cent grade. The rolling circumference of the loaded rear wheels is 8.96 ft., whence their diameter is 34.1 in. (nominally 34-in. tires). The friction loss from the cylinders to the tires is taken at 50 lb. For the particular car under discussion, let the

gear ratio (first) be $\frac{12}{1}$. Then assume that the motor will work with constant tractive force up to 1200 r.p.m., when the r.p.m. at the wheel will be $\frac{1200}{12} = 100$, and the feet per second

$$V_0 = \frac{34.1 \times 100\pi}{12 \times 60} = 14.83$$

and

$$V = \frac{14.83}{1.462} = 10.17$$

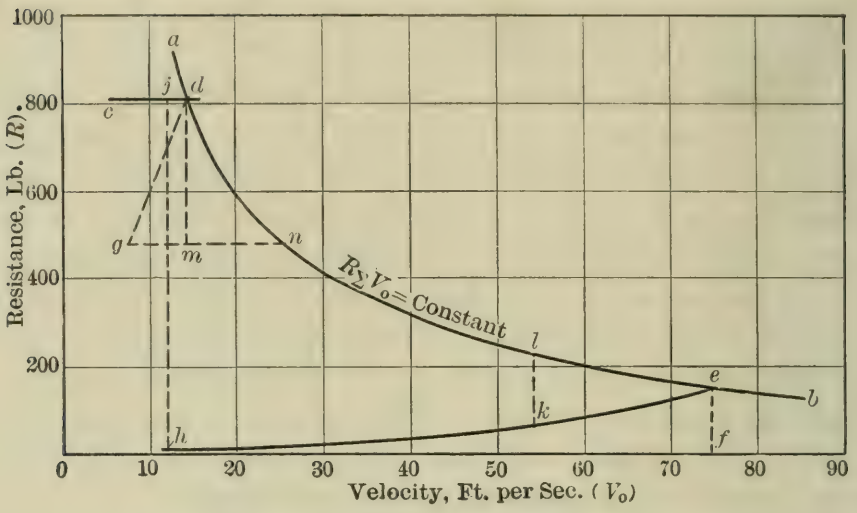


FIG. 3 HEAD-END AND ACCELERATING FORCES

At this speed, the motor will develop $28.25\left(\frac{162.5}{212.5}\right) = 21.6$ h.p. at the wheels, and therefore

$$R = \frac{375.5 \times 21.6}{10.17} = 802 \text{ lb.}$$

Then, since $dV_0 = 1.462\ dV$

$$\frac{dV}{dt} = \text{constant} = \frac{32.2 \times 802}{1.462 \times 3300} = 5.35$$

and for the attainment of a velocity of 5 miles per hour the elapsed time is $\frac{5}{5.35} = 0.94$ seconds. In the velocity-resistance diagram, Fig. 3, we therefore draw the horizontal line cd from $V_0 = 0$ to $V_0 = 14.83$, with $R = 802$. In the speed-time diagram, Fig. 4, we draw the straight line Oa from $V = 0$, $t = 0$ to $V = 5$, $t = 0.94$.

15 The point a in Fig. 4 lies between c and d in Fig. 3. To determine the speed-time relation beyond a (Fig. 4) and up to d (Fig. 3), we must consider both head-end resistance and acceleration resistance. The sum of these remaining constants

$$R_\Sigma = 802 = 0.0025 A V^2 + \frac{m}{32.2}\frac{dV_o}{dt}$$

$$= 0.065 V^2 + 102.3\,\frac{dV_o}{dt}$$

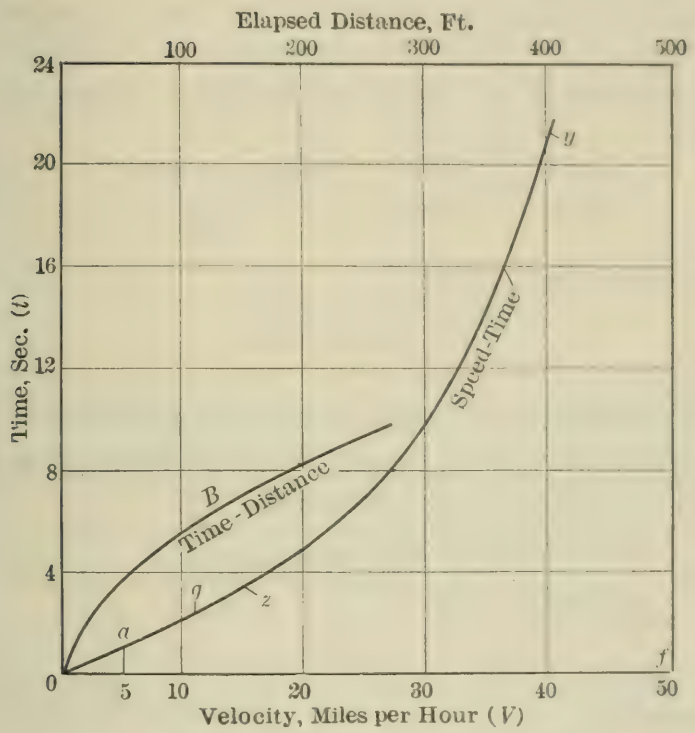


FIG. 4 ACCELERATION CURVES

This gives, for $V=5$, $t=0.94$; for $V=10$, $t=1.87$; and for $V=10.17$ (limit), $t=1.92$. The curve ag in Fig. 4 is plotted with these values.

16 From this point

$$\text{h.p.} = \text{constant}^* = \left(0.0025\, A V^2 + \frac{m}{32.2}\frac{dV_o}{dt}\right)\frac{V_o}{550}\dagger$$

* More strictly, as shown by equation [3], the horsepower does not increase quite as rapidly as the piston speed; but it seems a fair simplification of the problem to proceed in the manner suggested.

† The rate of doing work when the instantaneous force is R_Σ and the instantaneous velocity is V_o, is the same as that when a constant force R_Σ moves the car at the constant velocity V_o for one second of time.

If the car moves in a vacuum, so that there is no head-end resistance

$$\text{h.p.} = \text{constant} = \frac{m}{32.2}\,\frac{dV_0}{dt}\,\frac{V_0}{550}$$

the equation of a parabola having its vertex at the origin and its focus in the time axis. In the general case

$$R_\Sigma V_0 = \text{a constant}$$
$$R_\Sigma V = \text{a constant}$$

equations of an equilateral hyperbola referred to its asymptotes, the axes of R_Σ and V or V_0. This is plotted as the curve *adleb*, Fig. 3, and continues until the head-end resistance alone consumes the power, when

$$\text{h.p.} = \text{a constant} = \frac{0.0025\,A\,V^2\,V_0}{550}$$

which for our conditions gives

$$V = 50,\ V_0 = 72.1$$

point *e*, Fig. 3.

17 From *c* through *d* to *e*, Fig. 3, the total resistance is the sum of two items, which may be separately shown. Thus, at any velocity V_0 ($=1.462V$), the head-end resistance is

$$0.0025\ AV^2 = 0.0303\ V_0^2$$

from which we have

Velocity, ft. per sec.	10	20	30	40	50	60	70
Head-end resistance	3.03	12.1	27.3	48.5	75.9	109	148

These values give the curve *ohke*, Fig. 3. Any ordinate, like *hj*, *kl*, lying between this curve and *cde*, represents the force available for producing acceleration, at the given velocity.

18 The first expression in Par. 16 gives

$$dt = \frac{V\,dV}{54.2 - 0.000435V^3}$$

whence

$$t = 15.32\left[\frac{1}{2}\log_e\left(\frac{2500 + 50V + V^2}{2500 - 100V + V^2}\right) - 1.733\,\tan^{-1}\left(\frac{2V + 50}{86.5}\right)\right]$$
$$\pm \text{ a constant}$$

which yields

Velocity, miles per hour.....	10.17	11	15	20	30	40
Elapsed time from starting, seconds..	1.92	2.06	2.98	4.78	10.10	20.3

The curve is now plotted to the right from g, Fig. 4.

19 To express acceleration in terms of distance traversed, a graphical approximation from Fig. 4 should be sufficient. Since $dV = \frac{dS}{t}$, S, the distance traversed in the time t at the velocity V, is the area under the speed-time curve ogy. As far as the point z, where $V = 15$, it can be regarded as triangular; and the space, in feet, traversed from starting is, at a

$$\frac{5 \times 0.94 \times 1.462}{2} = 3.43$$

and at z

$$\frac{15 \times 2.98 \times 1.462}{2} = 32.6$$

From this point, accurate distances may be determined by a planimeter; or, approximately, from

$V = 15$ to $V = 20$, distance traversed $= 17.5 \times 1.80 \times 1.462 = 46.0$ feet, elapsed distance $= 32.6 + 46.0 = 78.6$ feet:

$V = 20$ to $V = 30$, distance traversed $= 25 \times 5.32 \times 1.462 = 194$ feet, elapsed distance $= 78.6 + 194 = 272.6$ feet.

These values give curve B, Fig. 4.

20 Considering the graph cde, Fig. 3: after the point d is passed, the piston speed (a function of V_0 at a given gear) becomes excessive. This is corrected by throwing in a new gear. If the car can for a short time maintain its velocity, the change of gear will decrease the r.p.m. and h.p. of the motor, and the graph from d will be as shown by the dotted lines dmn. If the car slows down, the action is as along $dqmn$. The curve de is then purely ideal, a sort of resultant of a series of steps like dmn or $dqmn$, and the degree of approach of the actual steps to the ideal curve will be related to the number of gear changes. Under a fixed maximum piston-speed limit, the velocity-resistance curve would consist of a series of alternate constant-velocity and constant-resistance paths, and the speed time curve would be made up of a series of alternate straight vertical lines and curves like ag, Fig. 4. Supernormal piston speeds introduce still further modifications of the ideal curve.

CONCLUSIONS

21 *a* The horsepower developed at the engine shaft by one single-acting 4-cycle cylinder at 1000 ft. piston speed may be written

$$\frac{Cd^2}{26000}$$

where d is the cylinder diameter in in. and C has a value between 5970 and 11008.

b The horsepower developed at any piston speed S may be written

$$\frac{Cd^2fS^e}{24{,}200{,}000}$$

tentative values for the constants being, $f=1.85$, $e=0.9$, $S>500$.

c The wheel-rim tractive force, in lb., necessary for speed only, on a level road, is

$$R_s=0.0025\ AV^2$$

A being the transverse clearance area of the car, in sq. ft., and V its speed in miles per hour.

d That necessary for a speed V_g on a grade k is

$$R_g=0.0025\ AV_g^2\ +\ mk$$

m being the weight of the car in lb.

e The tractive force referred to the engine shaft for a wheel-rim tractive force R is

$$R+g$$

g being probably about constant for a given car.

f The fundamental equation of design is then

$$\frac{Cd^2fS^e}{24{,}200{,}000}=\frac{(R+g)\,V}{375.5} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

which at 1000 ft. piston speed becomes

$$\frac{Cd^2}{69.1}=(R+g)\,V \quad \ldots\ldots\ldots\ldots\ldots\ldots [10]$$

g Equation [10] should be used to determine d at the condition of maximum power, and the corresponding necessary gear ratio is

$$u=\frac{336\,V}{Nw}$$

where N = r.p.m. of car wheel having a diameter of w in.

h For any other condition, independent assumptions as to velocity and horsepower are impossible. The engine must have sufficient speed to develop the required horsepower. If the wheel speed is also fixed, the gear ratio is fixed and the limit of tractive force established. The gear ratio for a given car is a function of both the total resistance and the piston speed.

No. 1338

ECONOMY OF 1000-KW. NATURAL GAS ENGINE

TESTS, CONSTRUCTION AND WORKING COSTS

By Edwin D. Dreyfus, East Pittsburgh, Pa.
Member of the Society
and
V. J. Hultquist,[1] Schenectady, N. Y.
Non-Member

A new power unit just placed in service at the Allegheny plant of the American Locomotive Company consists of a 1000-kw. generating set supplied with natural gas. The unit comprises a 23½ in. by 33 in. Westinghouse twin-tandem, double-acting, 4-cylinder engine, direct-connected to a 1000-kw. maximum rated direct-current generator, interpolar, compound-wound type of standard make. The gas for the plant is obtained from a 15-in. service main of the Manufacturers' Light and Heat Company, which is fed by their reducing station on the Brighton Road at Jacks Run.

2 An equipment of this kind represents about the simplest heat motor unit that may be installed, and is superior to the oil engine, as it requires no storage of fuel. An interior view of the plant is given in Fig. 1.

3 A hydraulic governing system is employed, the relays relieving the governor of heavy valve work. The cylinders have forced-feed lubrication, and are equipped with duplicate magnetic igniters. Three sources of ignition current are provided: (*a*) a motor generator, (*b*) a battery and (*c*) a lighting supply. The engine is so arranged that either half can be operated separately by removing the connecting rod of the other side, which presents advantages in making repairs, and by allowing economical operation when the shop load is under 500 kw. Gas is delivered to the pressure regulator through a 6-in. main at about 5 lb. pressure. Arrangement is made to adjust the engine readily for producer gas.

[1] Engineer-in-charge, American Locomotive Company.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

4 A switchboard of regular construction is installed, with connection to the old plant about 500 ft. distant. The station auxiliaries consist of two oil filters, two motor generators for ignition current, one air storage tank with a capacity of about 275 cu. ft. for starting. One single-stage, direct-connected air compressor of 14 ft. capacity, is used only for emergency at such times as the main plant system is not working. Two Bowser registering oil tanks assist in keeping a record of the oil consumption.

5 The muffler construction is worthy of special note. Due to the fact that the plant is located in a residential district and within 10 ft. of a dwelling, it became necessary to design a muffler that would completely obviate the objectionable noise of the exhaust. It is of monolith (reinforced concrete) construction, consisting of seven compartments, the two larger of which receive the exhaust from the main headers and are provided with relief valves for safety in case of after explosions from misfiring. The gases travel in a zigzag path.

6 The engine house is of steel and brick construction, with an engine room 40 ft. by 62 ft., the roof covering of reinforced-concrete tile and the floor of reinforced concrete. The floor is so constructed as not to come into contact with the engine foundations, in order to eliminate the transmission of vibration from the engine to the building foundation walls. The power plant was designed and installed under the direction of Wm. Dalton, chief engineer of the company.

SPECIFIED PERFORMANCE

7 The specifications stated that the engine must be capable of delivering a normal load of 1500 b.h.p. when operating at 150 r.p.m., with gas of not less than 950 effective B.t.u., and of developing 10 per cent overload for a period not exceeding 2 hours, and 20 per cent momentarily. The following efficiencies were named:

Full-load, 10,000 effective B.t.u. per b.h.p-hr.
¾-load, 11,200 effective B.t.u. per b.h.p-hr.
½-load, 13,300 effective B.t.u. per b.h.p-hr.

About 7 gal. of cooling water per b.h.p-hr. are required with cooling-water supply at 70 deg. fahr.

OBJECT OF TEST

8 When these tests were undertaken, they had for their object a twofold purpose: (*a*) to prove the guaranteed efficiencies, and (*b*)

to determine the influence of varying conditions on the performance with a view to acquainting the operating force with the essential features which affect both the economy and safety of operation. It was also desired that the usual adjustments and conditions which would be maintained in the regular operation of the engine be observed while the tests were in progress, and all refinements of test which would contribute towards realizing the highest obtainable efficiencies were intentionally neglected.

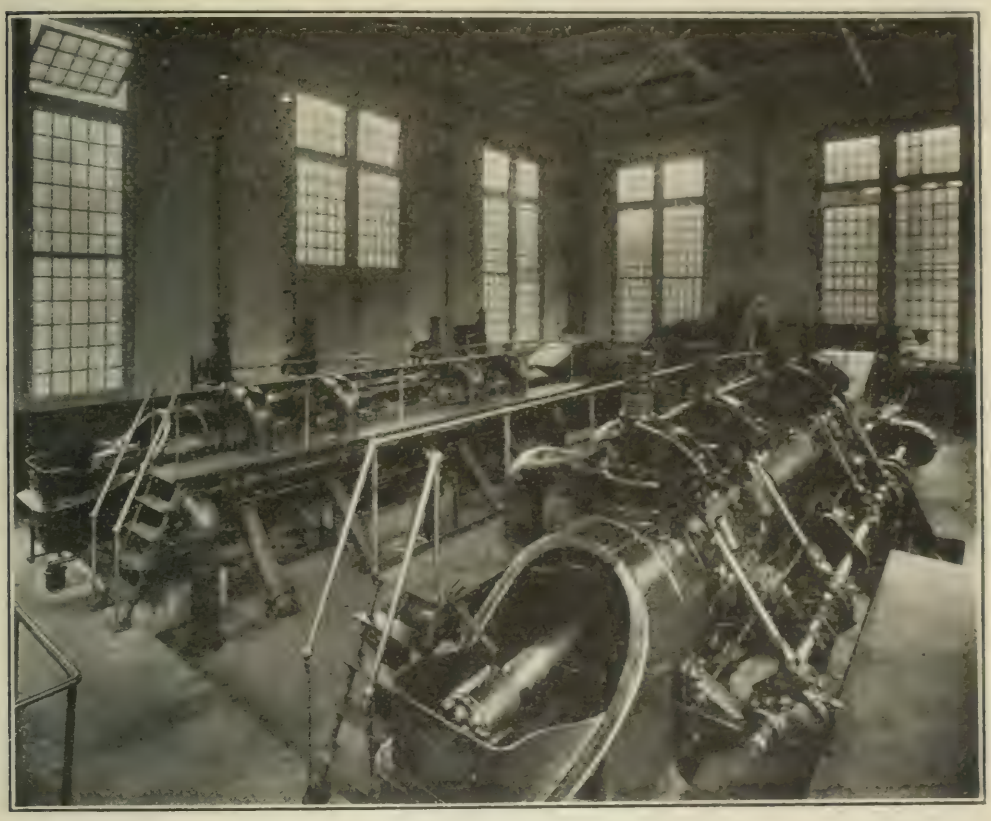


FIG. 1 INTERIOR VIEW OF THE PLANT

9 These tests would, therefore, establish results which engineers in charge of the power house could attain in ordinary operation, and be employed as a criterion for their average monthly consumption in order to ascertain whether or not the unit was in good order.

10 In addition to determining the heat economy of the engine, arrangements were made to obtain the following: (*a*) jacket water consumption; (*b*) heat balance; (*c*) the cylinder and lubricating oil consumption of the engine.

CONDITIONS OF TEST

11 As the engine was readily isolated from the remaining power equipment, it became a simple matter to provide for these tests. At the time they were conducted, the shop load was below normal and a water rheostat was therefore installed to supplement the demand in order to operate the engine at its normal and overload ratings. The rheostat afforded another advantage of regulating the load placed on the unit, as the cranes and motor-driven shop tools had a tendency to create a surging load.

12 The gas consumption of the engine was obtained with a Wylie proportional gas meter, which constituted a part of the regular station equipment. Calibration tests were made at the plant with an orifice prover and later verified by the "holder drop" method. These tests proved that the meter ranged from 15 to 20 per cent fast. Disk-type water meters (in duplicate) likewise afforded a means for obtaining the water consumption.

13 During several tests trouble was experienced, due to the line pressure at the meter falling very low, and at certain periods the draft of the engine created a suction in the gas header. At such times, some of the cylinders (left-hand No. 1 particularly) failed to receive the charge of mixture, which prejudiced the economy as the result of the idle operation of these cylinders.

SUMMARY OF RESULTS

14 *Capacity.* Unfortunately no maximum capacity test in excess of rating could be conducted, this being prevented by the inadequacy of the resistance box which was used, since it was impossible to secure adequate load as the plant was not running at full capacity. But an average load of 1057 kw. was thrown on the unit for 2 hours, and the condition of operation and position of the governor (the valve ports not being entirely open), were such that it seems apparent that a greater load could have been readily sustained.

15 *Economy.* Results of economies at various loads are given in Table 1 and graphically shown in Fig. 2. Owing to changes in power demand in the shop, the load could not be regulated to correspond with the guaranteed points. By using total heat quantities and plotting guaranteed consumption, there is practically a coincidence of the actual results and the rated efficiency of the engine.

16 *Gas.* The average heat value obtained during the test was 1055.3 (24 determinations with Junker calorimeter), varying from

1020 to 1066 effective B.t.u. (Table 7). The specific gravity by effusion test was 0.692 (see Table 9). This ranges very high in heat values, being about $8\frac{1}{4}$ per cent higher than the accepted average for natural gas (975 B.t.u.) upon which basis the gas engine was designed.

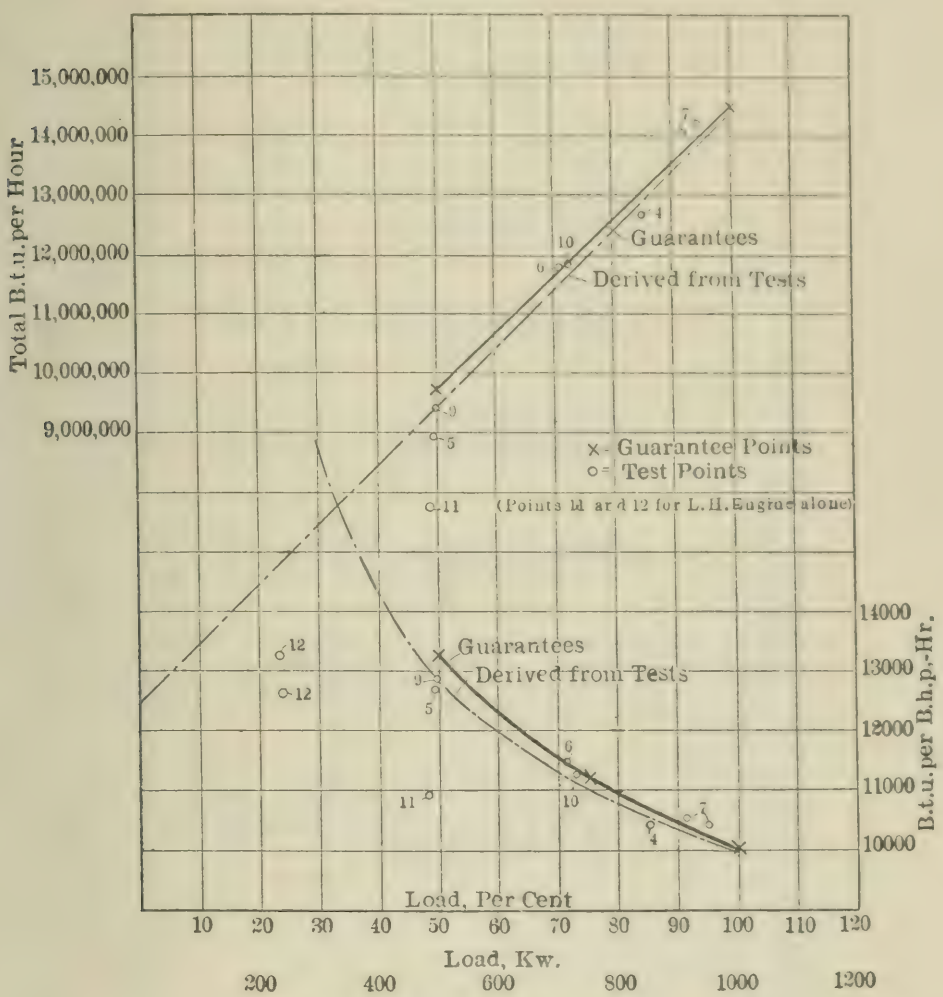


FIG. 2 ECONOMIES AND GUARANTEES, 1000-KW. ELECTRIC UNIT

17 *Combustion.* Combustion was regular when the gas supply was normal, and the indicator cards showed good burning, no backfiring or prematuring occurring during any of the runs. At short periods during some of the runs the gas supply was greatly reduced, owing to insufficient delivery pressure at the engine[1] and the cyl-

[1] The gas lines were new and dirt accumulated on the screen ahead of the regulator.

TABLE 1 SUMMARY OF TEST RESULTS, WESTINGHOUSE 1500-H.P. TWIN-TANDEM GAS ENGINE, AMERICAN LOCOMOTIVE COMPANY, ALLEGHENY, PA.

Run	4	5	6	7	8	9	10	11	12
Duration, hr	2½	2½	3	4	2	2½	2½	3½	1
Per Cent Rating, Engine	81.4	47.75	68.6	90.4	101.7	48.3	70.3	95.0	49.3
Per Cent Rating, Generator	85.1	49.1	71.6	94.6	105.7	49.7	72.8	48.8	24.4
Load kw. Corrected	851	491	716	946.5	1057	497	728	488	244
Load b.h.p. Corrected	1220	715	1028	1358	1520	724	1045	712	370
Heat Value Gas	1055	1055	1055	1032.5	1048.7	1053	1053	1065	1065
Gas per kw-hr., cu. ft., 62 deg. 30 in. Hg	14.4	17.28	15.65	14.52	14.22	17.78	15.40	14.91	18.07
B.t.u. per kw-hr	142915	18240	16510	14995	14915	18730	16215	15890	19250
Gas per b.h.p-hr. cu. ft. 62 deg. 30 in. Hg	9.88	11.85	10.88	10.11	9.94	12.1	10.68	10.24	12.42
B.t.u. per b.h.p-hr	10410	12510	11480	10450	10410	12850	11280	10910	13220
Thermal Efficiency, per cent b.h.p	24.45	20.35	22.22	24.34	24.45	19.8	22.55	23.3	19.28
Temperature Gas to Engine, deg	71.0	71.2	73.0	71.1	72.0	73.4	74	69.9	70
Cooling Water per b.h.p-hr., gal	8.57	11.25	8.71	7.54	6.88	12.54	8.86	7.03	13.95
Mechanical Efficiency, per cent	79.9	68.8	72.2	79.3	78.2	68.0	74.2	73 3	65 0
Heat to Jacket and Pistons, per cent	42.6	51.7	47.4	45.1	42.45	50.4	47.15	42.7	56.3
Exhaust and Radiation, per cent	32.95	27.95	30.40	30.56	33.1	29.8	30.3	34.0	24.42

Natural Gas, West Virginia.

TABLE 2 HEAT BALANCE. WESTINGHOUSE 1500-H.P. TWIN-TANDEM GAS ENGINE TEST, AMERICAN LOCOMOTIVE COMPANY, ALLEGHENY, PA.

Run	Generator Load, Kw.	Engine Load, b.h.p.	Water						Total Heat Supplied per b.h.p-hr.	Heat Equivalent of b.h.p-hr.	Jacket Absorption b.h.p-hr.	Heat Loss Exhaust by Balance	Engine		
			Gal. per hr., Total	Gal. per b.h.p.	Lb. per b.h.p.	Inlet Temperature	Discharge Temperature	Rise, Deg. Fahr.					Heat* Balance, Per Cent		
													Work, b.h.p.	Jackets	Exhaust and Radiation
4	851	1220	10460	8.57	71.4	75	137	62	10410	2545	4430	3435	24.45	42.60	32 95
5	491	715	8050	11.25	93.7	75	144	69	12510	2545	6465	3500	20.35	51.70	27.95
6	716	1028	8960	8.71	72.6	75	150	75	11480	2545	5445	3490	22.20	47.40	30.40
7	946.5	1358	10234	7.54	62.8	74	149	75	10450	2545	4710	3195	24.34	45.10	30.56
8	1057	1520	10420	6.88	57.4	72	149	77	10410	2545	4420	3445	24.45	42.45	33.1
9	497	724	9080	12.54	104.6	72	134	62	12850	2545	6480	3825	19.80	50.4	29.8
10	728	1045	9275	8.86	73.8	72	144	72	11280	2545	5315	3420	22.55	47.15	30.3
11	488	712	4920	7.03	58.6	71.5	151	79.5	10910	2545	4660	3705	23.30	42.7	34.0
12	244	370	4960	13.95	116.3	71	135	64	13220	2545	7445	3230	19.28	56.3	24.42
13	....	i.h.p. 273	4938	18.08	150.5	72	121	49	.....	....	....	....	.....	....	....

* See Par. 29.

Note:—Run No. 7, Results from morning run only.

inders did not all receive the proper charge of mixture, causing some to fire weakly, or occasionally not to fire at all.

18 *Water.* The cooling water entered the engine at a temperature of from 72 to 76 deg. fahr, and left the jackets at an average temperature of about 136 deg. and the pistons at an average temperature of about 156 deg. The outlet water temperature could have safely been maintained higher, probably with a beneficial influence on the economy of the engine, but it was desired to operate with

TABLE 3 I.H.P. AND MECHANICAL EFFICIENCY

Date	Run	Duration Hr.	R.P.M.	Average M.E.P.	I.H.P.	Load Kw.	B.H.P.	Mechanical Efficiency
6-14-11.....	4	2½	149.3	77.76	1526.5	851.0	1220	79.9
6-14-11.....	5	2 1/2	150.5	52.45	1039.8	491.0	715	68.8
6-14-11.....	6	3	149.3	72.26	1422.1	716.0	1028	72.2
6-15-11.....	7	4	148.3	88.04	1712.3	946.5	1358	79.3
6-16-11.....	8	2	147.8	100.0	1947.8	1057.0	1520	78.2
6-16-11.....	9	2½	150.3	53.75	1064.6	497.0	724	68.0
6-16-11.....	10	2½	149.1	71.84	1409.8	728.0	1045	74.2
6-17-11.....	11†	3½	148.6	99.26	972.0	488.0	712	73.3
6-17-11.....	12†	1	150.7	57.77	569.1	244.0	370	65.0
6-20-11.....	13‡	1 5/6	152.6	27.17	273.0	Friction		

* Not corrected for errors in spring scale and generator efficiency, they being indeterminate and estimated to amount to 5 per cent, or 5 per cent total.

† Left-hand engine alone (right-hand engine disconnected).

‡ Right-hand engine not firing. Left-hand engine carrying friction load.

extreme conservatism, even during these tests, so that the practice established could be followed by an operating force of limited gas-engine experience.

19 The water consumption varied fron 6.88 gal. per b.h.p-hr. at overloads, to 12.54 gal. per b.h.p-hr. at ½-load (Table 1). No regulation was made on light loads for operating reasons, on account of a possible heavy increase in load such as might take place in regular service.

20 *Oil.* The quantity of oil (standard gas-engine oil for bearings and "Diamond A" for cylinders) average 0.68 gal. per hr. For the bearings the consumption was 0.256 gal. per hr. and for the cylinders 0.424 gal. per hr.

21 *Mean Effective Pressure and Mechanical Efficiency.* The mean effective pressure was computed from average cards selected through-

out each run. The mechanical efficiency of the engine thus obtained (about 80 per cent at full-load) is considerably less than the average. This 5 per cent discrepancy may be explained by errors in the generator efficiencies and indicator springs, the calibrations of which were inadvertently lost. The generator field temperature rise was 95 deg. at full-load.

22 *Regulation.* The average speed of the engine at full-load was 147.8 r.p.m., and at no-load, 152.6 r.p.m., giving a maximum varia-

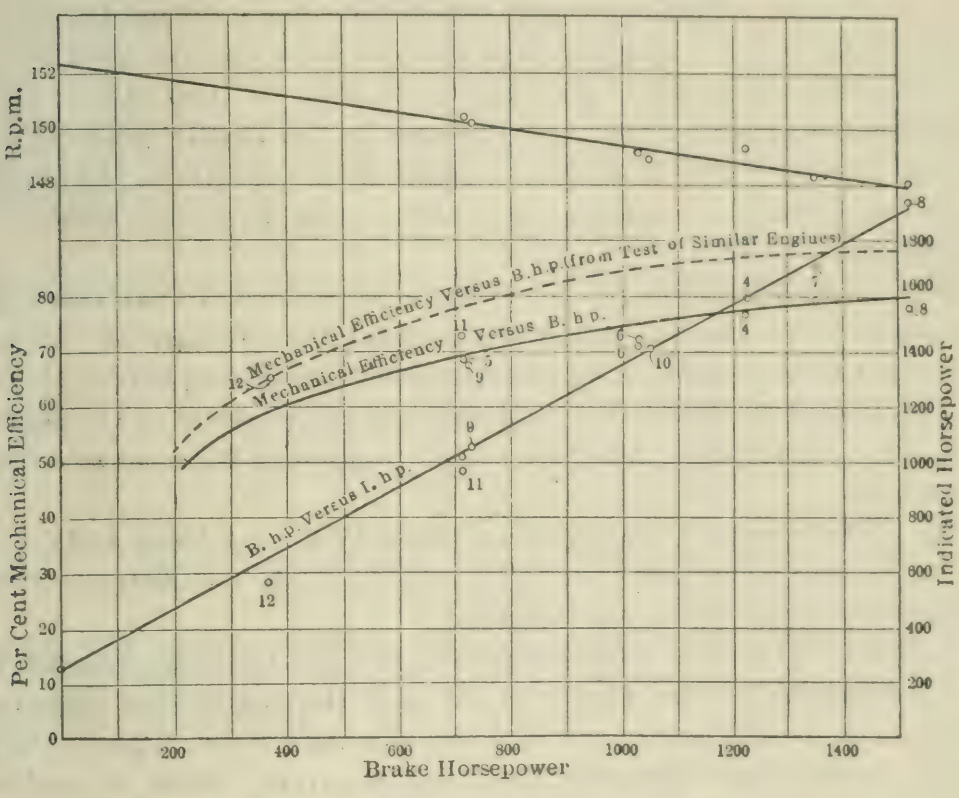


FIG. 3 DIAGRAM SHOWING MECHANICAL EFFICIENCY

tion of 3.1 per cent. Guarantees were 3 per cent plus and minus mean speed. The regulation curve, Fig. 3, is practically a straight line and shows good port design and governor action.

DEDUCTIONS FROM OBSERVATIONS

23 Inasmuch as there are invariably modifying conditions which influence the results recorded during tests of this nature, it is believed that it is not only warrantable but urgent that a fair interpretation be made of data obtained and properly reduced to consistent values.

Hence the following comment on the weight to be given to the various observations, is appended, and the typical performance line (shown by the dot and dash line, Fig. 2) is described.

24 It is to be noted that this engine is designed to be used on producer gas in the event of failure of natural gas. It is capable of sustaining its maximum load on gas of heat value as low as 950 B.t.u., and the ports are, therefore, large enough to admit the required amount of gas to the combustion chamber under these conditions. When the gas reached its maximum heat value, the engine showing fell off slightly. This is brought out in tests made during the afternoon of run No. 7. This was best with the heat value at 1042 B.t.u. and less favorable when the heat value rose to 1056 B.t.u. The gasometer was set so that the pressure in the supply pipes to the engine ranged from $\frac{1}{4}$ in. to $\frac{3}{4}$ in. Under these conditions, the engine pulled too strongly on gas and the mixture was, therefore, somewhat inefficient.

25 It would, therefore, prove more economical to set both the reducing valve and the gasometer so that the pressure of the gas delivered to the engine would approximate one atmosphere. It will be observed from Test No. 4 that when the gas pressure to the engine fell below atmosphere, the guaranteed economies were considerably bettered.

26 This curiously happened accidentally during these tests, but it could have been provided for specially, as in the Kenova tests[1] where this same effect was recorded.

27 After the tests were computed, the gas butterfly damper in the gas manifolds were partly closed, and the engine was observed to keep up to load and speed with this partially reduced gas supply.

28 Aside from what has already been given, there is another contributing factor which has apparently made the test points (referred to the brake horsepower output) higher than they should actually be as compared with the guaranteed performance; viz., the question of true efficiency of the generator.

29 The brake horsepower efficiencies deduced in this report are based upon the builder's rated efficiencies of the generator, given in Fig. 4. Efficiencies of generator, as ordinarily specified, do not include the load losses, and the temperature rise was abnormal[2] in actual operation, indicating a loss of energy from this cause.

30 Therefore, in rationalizing the various test points, due cog-

[1] Similar type of natural gas engine.

[2] Due to too wide air gap. This was afterwards remedied.

nizance has been taken of the different factors and working conditions which have already been described, and the representative performance has been derived as shown by the dot and dash total heat line and economy curve in Fig. 2.

31 From the improvement shown by the test points 11 and 12, it is to be recommended that one-half the engine be operated when low load prevails regularly.

32 It is evident that in these tests the water consumption was normal; however, water was used freely on light loads.

33 Oil consumption data collected during the tests were found to agree fairly in amount of consumption with gas engines of similar design, although owing to the fact that sufficient time had not elapsed for the engine to wear to good bearing, oil was used liberally and may presently be reduced.

34 Gas measurements were reduced to standard conditions, 62 deg., 30 in., by assuming a dry gas. If there was a small percentage

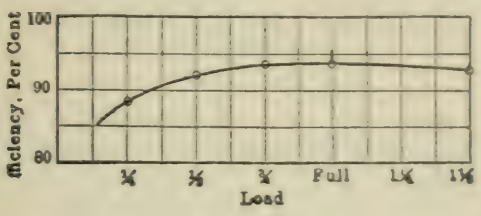


FIG. 4 PROBABLE GENERATOR EFFICIENCY CURVE

of gasolene vapors present, as is likely with this rich gas in certain localities, it may have lowered the gas consumption by a small fraction, as noted in similar tests.

CONSTRUCTION COSTS

35 As is evident from Table 5, interesting economies may be achieved in installing this type of equipment, the customary producer or equivalent boiler expense being eliminated.

36 It can hardly be said that these data furnish typical installation costs, as many of the items were subject to local factors and a part of the installation was made in the coldest winter weather. Nevertheless, they serve as a fair guide for general purposes, and the variations which might take place would represent a small percentage, as fixed charges, of total cost of the unit power. The cost of larger installations may be reduced somewhat, but that of smaller plants will be increased appreciably. The figures make rather a notable comparison with those for modern steam stations, which for similar

TABLE 4 COMPLETE COST OF POWER, 1000-KW. NATURAL GAS ENGINE

	22-Hr. Day						10-Hr. Day							
	Full Load		Three-quarters Load		One-half Load		Full Load		Three-quarters Load		One-half Load		One-half Engine Operating Full Load	
	Per Day	Per Kw-Hr.	Per Day	Per Kw-Hr.	Per Day	Per Kw-Hr.	Per Day	Per Kw-Hr.	Per Day	Per Kw-Hr.	Per Day	Per Kw-Hr.	Per Day	Per Kw-Hr.
Kw. Output	22000		16500		11000		10000		7500		5000		5000	
Load Factor, Per Cent	90.2		67.65		45.1		41.7		31.3		20.9		20.9	
Labor and Supervision	$10.75	$0.048	$10.75	$0.065	$10.75	$0.098	$5.75	$0.057	$5.75	$0.076	$5.75	$0.115	$3.75	$0.075
Gas at 14 cents per 1000 Cu.Ft.	41.21	0.19	33.33	0.202	24.75	0.225	19.00	0.190	15.15	0.202	11,25	0.225	9.60	0.192
Water at 10 cents per 1000 Gal.	24.20	0.11	19.30	0.117	17.49	0.159	11.00	0.110	8.77	0.118	7.94	0.159	4.70	0.094
Oil, Grease and Waste	5.45	0.025	5.45	0.033	5.45	0.049	2.48	0.025	2.48	0.033	2.48	0.049	1.24	0.025
Repairs and Maintenance	2.93	0.012	2.81	0.017	2.64	0.024	2.58	0.026	2.48	0.033	2.35	0.047	2.35	0.047
Total Disbursements per day	84.54	0.385	71.64	0.434	61.08	0.555	40.81	0.408	34.63	0.462	29.77	0 595	21.64	0.433
Fixed Charges*	33.40	0.151	33.40	0.203	33.40	0.304	33.40	0.334	33.40	0.445	33.40	0.668	33.40	0.668
Total	117.94	0.536	105.04	0.637	94.48	0 859	74.21	0.742	68.03	0.907	63.17	1.263	55.04	1.101

* Including interest, depreciation, insurance and taxes, 12½ per cent of original investment, $80,180.

work would cost from 100 to $130 per kw. installed. Data of this nature are required in the crucial determination of the most economical source of power.

TABLE 5 CONSTRUCTION COST 1000-KW. GAS-ENGINE INSTALLATION

	Per kw.
Land	$3.44
Building and Foundations	10.54
Engine Accessories and Foundations	41.25
Generator Erected	10.50
Piping and Storage Tanks	2.38
Stacks and Flues	.21
Muffler	.69
Switchboard	2.71
Station Wiring, Motors and Batteries*	5.96
Engineering	2.50
Total	$80.18

* Includes connecting cable between power stations.

COST OF PRODUCTION

37 The real measure of the value of any particular equipment lies in its final cost of operation when all accounts are taken into consideration. From the preceding data and other plant records, Table 4 has been prepared to show the complete cost to the company of generating power at various loadings of the plant. A distinction is made between 10-hr. and 22-hr. power, as industrial plants vary in the number of working hours per day. The use of 22 hours may not be followed generally, but it is the custom at this plant of the American Locomotive Company to have 1 hour's cessation of work between each 12-hr. shift.

38 On the B.t.u. basis , the gas delivered to the engine at 14 cents is equivalent in cost to coal of 12,500 B.t.u. at $3.50 per short ton. Consequently a considerable rise in the price of gas may be endured without the economic value of the plant suffering.

39 It will be noted that owing to the purchase of city water at a rate of 10 cents per 1000 gal., this item amounts to approximately 25 per cent of the operating cost. The company is considering the installation of a pumping station for their works, and it is estimated that the cost of water will be reduced to about 3 cents per 1000 gal., resulting in lowering the operating cost about 17½ per cent.

40 The plant has not been operated long enough to obtain posi-

tive data on repairs and maintenance and losses from decline in efficiency. In order that the values given in Table 4 may be conservative, a liberal allowance has been introduced for these factors, graded from 1.1 per cent of the investment value of the plant at full-load to 0.8 per cent at half-load, and distributed over the entire year, as determined in corresponding cases. For the slight change in efficiency which may be caused by leaks, cutting in and cutting out of service, etc., 3 per cent full-load to 5 per cent at half-load is allowed.

41 These economies are evidently very good and are much lower than current could be purchased at wholesale from large supply stations. Advantage of this type of equipment can only be taken in or adjacent to the natural gas belts where commercial pipe lines are laid.

42 It is important to note that no standby losses are met with in this type of plant, and it may be placed on the line in less than ½ hour after notice for power is given.

TABLE 6 GAS MEASUREMENTS, AMERICAN LOCOMOTIVE COMPANY ALLEGHENY, PA.

Run	Duration Hr.	Kw.	B.H.P.	Total Hourly Consumption as Measured, Cu. Ft.*	Barometer	Pressure Gas in H_2O	Temperature Deg. Fahr.	Meter Coefficient	Corrected Cu. Ft. Standard Gas per Hr., 62 Deg., 30 In.
4	2½	851	1220	15592	28.9	0.07	71	0.815	12030
5	2½	491	715	10114	28.9	0.87	73	0.862	8485
6	3	716	1028	13977	28.9	0.83	73	0.824	11200
7	4	946.5	1358	17534	29.0	0.75	73	0.806	13740
8	2	1057	1520	19345	29.0	0.66	72	0.802	15040
9	2½	497	724	10692	29.0	0.66	73	0.855	8840
10	2½	728	1045	14084	29.0	0.66	74	0.823	11210
11	3½	488	712	8436	29.05	0.66	70	0.885	7280
12	1	244	870	4790	29.05	0.66	70	0.945	4410

* NOTE: Determined by plotting all 15 minute readings, and examining for errors.

METHOD OF TESTING

GAS MEASUREMENTS

43 For the quantity of gas consumed by the engine, a 6-in. Wylie proportional meter was depended upon, and its actual readings corrected in accordance with calibrations made in two different ways. It is known, of course, that this type of meter must be used with every precaution in order that the results may prove reliable. The meter was located about 14 ft. from the butterfly damper and 28 ft. from the branch pipes to each side of the engine. Building dimension limits prevented installing the meter at any greater distance, which is probably better practice, owing to the fact that it admits of consider-

able receiver capacity which would dampen the slight fluctuating of pressure in the connecting pipes. What real effect this would have on the meter is more or less indefinite, but it is probable that any rapid changes in flow would allow the regulating valve to remain near to its seat, causing great drop across the meter, and consequently it would have a tendency to register fast under these conditions.

44 However, the main pipe, 28 ft. in length, leading from the meter to the engine, is of large diameter (12 in.), having been made of ample capacity to allow for operation on producer gas. The storage capacity thus introduced served a double purpose in dampening pressure fluctuations and in creating a negligible pressure loss, so that any inaccuracies from these causes are believed to be of no moment. In fact, the pressure immediately beyond the meter did not vary perceptibly.

45 A tee-connection was placed in the line beyond the meter and an orifice prover provided, being used as shown in Fig. 5. Pressures were maintained in

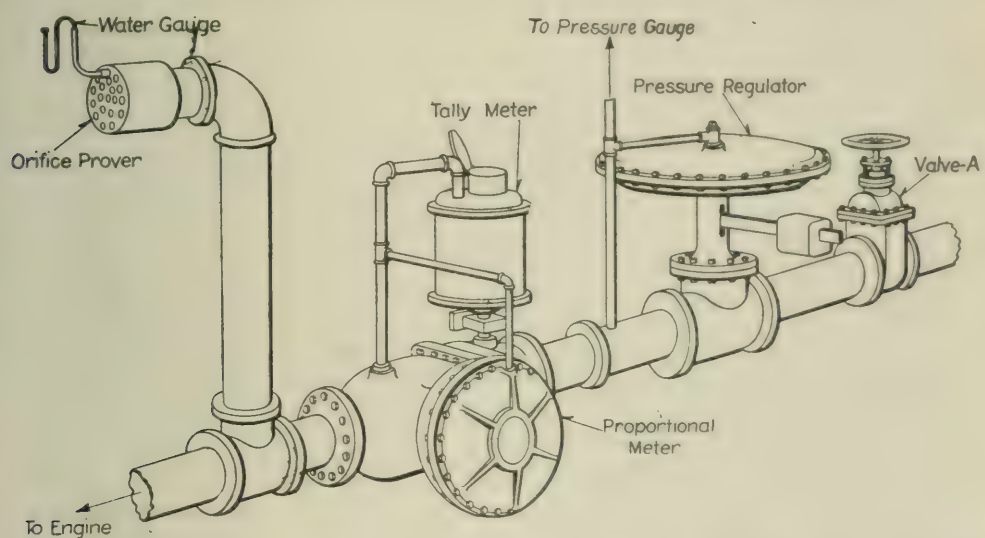


FIG. 5 DIAGRAM OF CONNECTIONS FOR PROVING METER BY STANDARD ORIFICES. PROVER REMOVED DURING PROGRESS OF TESTS AND CONNECTION SEALED BY BLANK FLANGE

the prover for different gas temperatures, specific gravity and barometric readings. The specific gravity was determined with an effusion flask, the readings from which averaged 0.692. The prover consists of a cylindrical vessel, the head of which is pierced with orifices designed to pass a known volume of gas when a definite pressure is maintained within it, this pressure depending upon barometric conditions, the temperature of the gas and its specific gravity (see Fig. 6). With the exercise of ordinary care, the limit of error with this method should be less than 1 per cent.

46 In checking the meter by the holder drop, air is used and it therefore becomes necessary to determine the equivalent correction coefficient for gas at a given rate of flow by dividing the air measurements by the ratio of the square of the relative densities (see Fig. 10).

47 Barometric pressures were obtained from the Weather Bureau at Pittsburgh, which were used directly without correction for difference in altitude,

Bar.	28.6	28.7	28.8	28.9	29.0	29.1	29.2	29.3	29.4	29.5	29.6	29.7	29.8	29.9	30.0
TEMP															
32	4.22	4.24	4.26	4.27	4.28	4.30	4.31	4.33	4.34	4.36	4.38	4.39	4.40	4.42	4.43
33	4.22	4.23	4.24	4.26	4.27	4.29	4.30	4.32	4.33	4.35	4.36	4.38	4.39	4.40	4.42
34	4.21	4.22	4.24	4.25	4.26	4.28	4.29	4.31	4.32	4.34	4.35	4.37	4.38	4.39	4.41
35	4.20	4.21	4.23	4.24	4.26	4.27	4.29	4.30	4.31	4.33	4.34	4.36	4.37	4.39	4.40
36	4.19	4.20	4.22	4.23	4.25	4.26	4.28	4.29	4.31	4.32	4.34	4.35	4.37	4.38	4.39
37	4.18	4.19	4.21	4.23	4.24	4.25	4.27	4.29	4.30	4.31	4.32	4.34	4.35	4.37	4.38
38	4.17	4.19	4.20	4.21	4.23	4.24	4.26	4.27	4.29	4.30	4.31	4.33	4.34	4.36	4.37
39	4.16	4.18	4.19	4.21	4.22	4.24	4.25	4.26	4.28	4.29	4.31	4.32	4.34	4.35	4.37
40	4.15	4.17	4.18	4.20	4.21	4.23	4.24	4.26	4.27	4.29	4.30	4.31	4.33	4.34	4.36
41	4.14	4.16	4.17	4.19	4.20	4.22	4.23	4.25	4.26	4.28	4.29	4.30	4.32	4.33	4.35
42	4.13	4.15	4.16	4.18	4.19	4.21	4.23	4.24	4.25	4.26	4.28	4.29	4.31	4.33	4.34
43	4.13	4.14	4.16	4.17	4.19	4.20	4.21	4.23	4.24	4.26	4.27	4.29	4.30	4.31	4.33
44	4.12	4.13	4.15	4.16	4.18	4.19	4.21	4.22	4.24	4.25	4.26	4.28	4.29	4.31	4.33
45	4.11	4.13	4.14	4.16	4.17	4.19	4.20	4.21	4.23	4.24	4.26	4.27	4.29	4.30	4.31
46	4.10	4.12	4.13	4.15	4.16	4.18	4.19	4.20	4.22	4.23	4.25	4.26	4.28	4.29	4.30
47	4.09	4.11	4.12	4.14	4.15	4.16	4.18	4.19	4.21	4.23	4.24	4.25	4.27	4.28	4.29
48	4.08	4.10	4.11	4.13	4.14	4.16	4.17	4.19	4.20	4.21	4.23	4.24	4.26	4.27	4.29
49	4.08	4.09	4.11	4.12	4.13	4.15	4.16	4.18	4.19	4.21	4.22	4.24	4.25	4.26	4.27
50	4.07	4.08	4.10	4.11	4.13	4.14	4.16	4.17	4.19	4.20	4.21	4.23	4.24	4.26	4.27
51	4.06	4.08	4.09	4.11	4.12	4.13	4.15	4.16	4.18	4.19	4.20	4.22	4.23	4.25	4.26
52	4.06	4.07	4.08	4.10	4.11	4.12	4.14	4.15	4.17	4.18	4.19	4.21	4.23	4.24	4.25
53	4.05	4.06	4.07	4.09	4.10	4.12	4.13	4.14	4.16	4.17	4.19	4.20	4.21	4.23	4.24
54	4.04	4.05	4.06	4.08	4.09	4.11	4.12	4.13	4.15	4.16	4.18	4.19	4.21	4.23	4.24
55	4.03	4.04	4.06	4.07	4.08	4.10	4.11	4.13	4.14	4.16	4.17	4.19	4.20	4.21	4.23
56	4.02	4.03	4.05	4.06	4.08	4.09	4.11	4.12	4.13	4.15	4.16	4.18	4.19	4.21	4.22
57	4.01	4.03	4.04	4.06	4.07	4.08	4.10	4.11	4.13	4.14	4.15	4.17	4.18	4.20	4.21
58	4.00	4.02	4.03	4.05	4.06	4.08	4.09	4.11	4.12	4.13	4.15	4.16	4.17	4.19	4.20
59	3.99	4.00	4.02	4.03	4.04	4.06	4.07	4.09	4.10	4.11	4.13	4.14	4.16	4.17	4.19
60	3.99	4.01	4.02	4.03	4.04	4.06	4.07	4.08	4.10	4.12	4.13	4.14	4.16	4.17	4.18
61	3.98	4.00	4.01	4.02	4.04	4.05	4.07	4.08	4.10	4.11	4.12	4.14	4.15	4.16	4.17
62	3.97	3.99	4.00	4.02	4.03	4.04	4.06	4.07	4.08	4.10	4.11	4.13	4.14	4.15	4.16
63	3.97	3.98	3.99	4.00	4.02	4.04	4.05	4.06	4.08	4.09	4.11	4.12	4.14	4.15	4.16
64	3.96	3.97	3.99	4.00	4.01	4.03	4.04	4.06	4.07	4.09	4.10	4.11	4.12	4.14	4.15
65	3.95	3.96	3.98	3.99	4.01	4.02	4.04	4.05	4.06	4.08	4.09	4.10	4.12	4.13	4.15
66	3.94	3.96	3.97	3.99	4.00	4.01	4.03	4.04	4.05	4.07	4.08	4.09	4.10	4.11	4.13
67	3.93	3.95	3.96	3.98	3.99	4.00	4.02	4.03	4.05	4.07	4.08	4.09	4.10	4.11	4.12
68	3.93	3.94	3.96	3.97	3.98	4.00	4.01	4.03	4.04	4.05	4.07	4.08	4.09	4.11	4.12
69	3.92	3.93	3.95	3.96	3.98	3.99	4.00	4.02	4.03	4.05	4.06	4.07	4.09	4.10	4.11
70	3.91	3.92	3.94	3.96	3.97	3.98	4.00	4.01	4.02	4.04	4.06	4.07	4.07	4.08	4.10
71	3.91	3.92	3.93	3.95	3.96	3.98	3.99	4.00	4.02	4.03	4.05	4.05	4.08	4.07	4.09
72	3.91	3.91	3.93	3.94	3.96	3.97	3.99	4.00	4.01	4.01	4.02	4.03	4.05	4.06	4.08
73	3.90	3.91	3.92	3.94	3.95	3.96	3.98	3.99	4.00	4.01	4.02	4.03	4.05	4.06	4.08
74	3.90	3.90	3.91	3.93	3.94	3.95	3.97	3.98	4.00	4.01	4.02	4.03	4.05	4.06	4.07
75	3.88	3.90	3.90	3.92	3.93	3.95	3.96	3.97	3.98	3.99	4.00	4.02	4.03	4.04	4.06
76	3.87	3.88	3.89	3.90	3.93	3.94	3.95	3.96	3.97	3.99	4.00	4.01	4.03	4.04	4.06
77	3.86	3.88	3.89	3.90	3.92	3.93	3.95	3.96	3.97	3.99	4.01	4.01	4.03	4.04	4.06
78	3.86	3.87	3.88	3.89	3.91	3.92	3.94	3.95	3.97	3.98	3.99	4.01	4.02	4.03	4.05
79	3.85	3.86	3.88	3.89	3.90	3.92	3.93	3.94	3.96	3.97	3.98	4.00	4.01	4.02	4.04
80	3.84	3.85	3.87	3.89	3.90	3.91	3.92	3.94	3.95	3.96	3.98	3.99	4.00	4.01	4.03
81	3.84	3.85	3.86	3.88	3.89	3.90	3.91	3.93	3.94	3.96	3.98	3.99	4.00	4.01	4.02
82	3.83	3.84	3.85	3.87	3.88	3.90	3.90	3.92	3.93	3.95	3.96	3.98	3.99	4.00	4.01
83	3.82	3.84	3.85	3.86	3.88	3.89	3.90	3.91	3.93	3.94	3.96	3.97	3.98	3.99	4.0
84	3.82	3.83	3.85	3.86	3.88	3.89	3.90	3.91	3.93	3.94	3.96	3.97	3.97	3.99	4.00
85	3.81	3.82	3.84	3.85	3.86	3.88	3.88	3.90	3.91	3.92	3.94	3.95	3.96	3.98	3.99
86	3.80	3.81	3.83	3.84	3.85	3.87	3.88	3.89	3.90	3.92	3.93	3.95	3.96	3.97	3.99
87	3.80	3.80	3.82	3.83	3.84	3.86	3.87	3.88	3.89	3.90	3.92	3.94	3.95	3.97	3.98
88	3.79	3.80	3.81	3.83	3.84	3.85	3.87	3.88	3.89	3.90	3.91	3.93	3.95	3.96	3.97
89	3.78	3.80	3.81	3.82	3.84	3.84	3.86	3.87	3.88	3.89	3.91	3.92	3.94	3.95	3.97
90	3.77	3.79	3.80	3.82	3.83	3.84	3.85	3.86	3.87	3.88	3.90	3.91	3.93	3.95	3.96
91	3.76	3.78	3.80	3.81	3.82	3.83	3.84	3.86	3.87	3.88	3.90	3.91	3.92	3.94	3.96
92	3.76	3.77	3.79	3.81	3.82	3.82	3.83	3.85	3.86	3.87	3.89	3.90	3.91	3.93	3.94
93	3.75	3.76	3.78	3.79	3.80	3.82	3.83	3.84	3.86	3.87	3.88	3.89	3.90	3.92	3.93
94	3.75	3.76	3.77	3.78	3.79	3.81	3.82	3.84	3.85	3.86	3.87	3.88	3.90	3.91	3.92
95	3.74	3.75	3.76	3.78	3.79	3.81	3.82	3.83	3.84	3.85	3.86	3.88	3.89	3.91	3.92

FIG. 6 INCHES OF WATER PRESSURE REQUIRED TO PASS 1 FT. OF AIR PER SECOND THROUGH 1½-IN. ORIFICE. IN MEASURING GAS MULTIPLY THE PRESSURES BY SPECIFIC GRAVITY OF GAS

this being too slight to have any noticeable effect, only a small fraction of 1 per cent. Temperatures were taken after the gas passed the reducing valve,

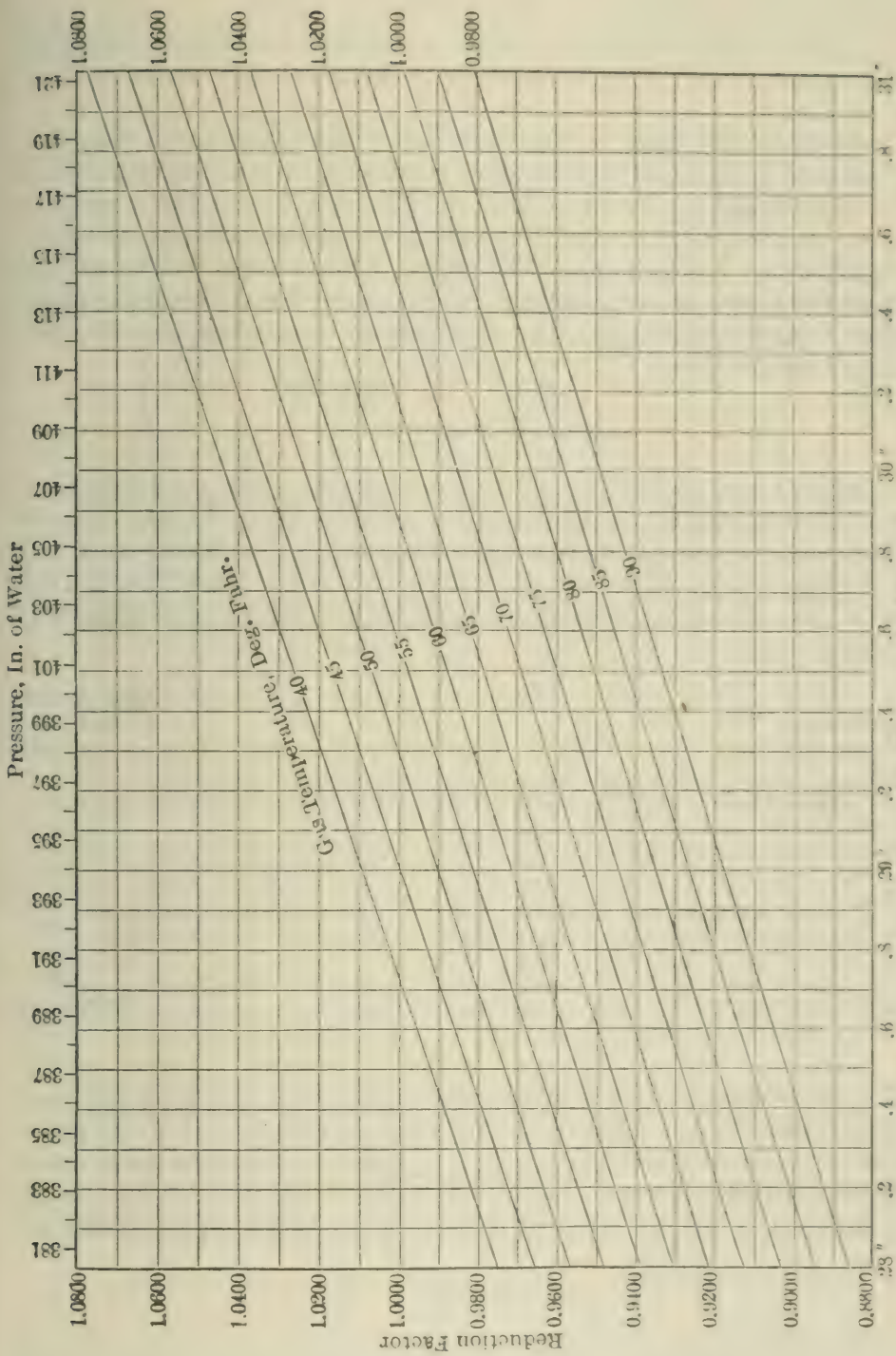


FIG. 7 CHART FOR REDUCTION OF GAS MEASUREMENTS TO STANDARD CONDITIONS, 62 DEG. FAHR., 30 IN. BAROMETER

which gives the true temperature of the gas as it passed through the meter. Gas pressures were taken at a point between the reducing valve and the meter. These readings were taken every few minutes and then plotted so that any irregularities could be noted.

48 The gas as measured was corrected to standard conditions, namely 30 in. mercury pressure and 62 deg. temperature, and a convenient reduction chart is given in Fig. 7.

49 Corrections were made on the basis of dry gas. This gas is very rich and carries with it considerable napthaline vapors, the presence of which would have the effect of making the correction factor greater, but hardly more than 1 per cent. As there were no means of making any determinations for conditions, no allowance has therefore, been made. Observations of all tests are plotted on log sheets (not published, but available for reference), which enable one to ascertain whether the observations varied consistently, one with respect to the other.

CHEMICAL ANALYSES

50 Provisions were made to obtain average samples of the gas during each run with suitable retainers. The samples, unfortunately, became diluted before satisfactory analyses were made. According to the heat value and specific gravity, it must closely approach the constituency determined for the Kenova, W. Va., gas, which is considered of enough value to be included and which is as follows:

Constituents	Composition, Per Cent
Carbon dioxide, CO_2	0.35
Oxygen, O_2	0.5
Illuminants, $\frac{1}{4}$ C_6H_6, $\frac{3}{4}$ C_2H_4	0.7
Carbon monoxide, CO	0.45
Hydrogen, H_2	0.0
Methane, CH_4	78.5
Ethane, C_2H_6	19.2
Nitrogen, N_2	0.3
	100.0
Net heat value, B.t.u. per cu. ft.	1049.6 B.t.u
Standard gas, average specific gravity	.6598

51 The exhaust gases were analyzed, the object being to test for the completeness of combustion. The CO content was a very small percentage on all loads except near rating and overloads. The higher CO on these loads shows that the gas butterfly valves should have been throttled for such high value gas; the engine drew gas too freely in proportion to the air supply (Table 8)

CALORIMETRY

52 Besides the chemical analyses, calorimeter determinations were made, with the results given in Table 7. A Junker calorimeter in the best of repair was employed and operated by a skilled observer. The instrument was located in the basement free from disturbances from draft or heat radiation. There

was a change of over 4 per cent variation in heat value during these tests, which is possibly accounted for by the difference in napthaline vapor content in the gas. Readings were taken about every 15 minutes so that the effect of this variation might be studied. This consideration has been taken up more fully in Par. 24.

TABLE 7 JUNKER CALORIMETER TEST OF NATURAL GAS (W. VA.)

Time	Pressure Gas at Meter Hg. In.	Temperature, Deg. Cent.			Water		Cu. Ft. Gas Used	B.t.u. Per Cu. Ft.		
		Gas at Meter	Average Outlet Water	Average Inlet Water	Average Rise in Temperature	Cooling cc.	Condensed cc.		Total	Effective	
										Not Corrected	Corrected to 30 In. Hg. 62 Deg. Fahr.
	H	*J*			*A*	*B*	*C*	*D*	*E*	*F*	*G*
6-15-11											
11.40 a.m.	0.01	25.0	36.8	23.6	13.23	1840	3.6	0.09	1075	990	1045
11.55 a.m.	0.01	25.0	34.9	23.6	11.37	1860	3.1	0.08	1050	967	1020
1.30 p.m.	0.01	25.0	37.5	23.5	13.73	1750	3 8	0.09	1060	970	1022
2.00 p.m.	0.01	25.0	38.0	23.5	14.53	1885	4.0	0.1	1089	1004	1059
2.15 p.m.	0.01	25.0	35.3	23.5	11.83	1860	3.2	0.08	1092	1007	1061
2.30 p.m.	0.01	25.0	35.4	23.5	11.9	1835	3.0	0.08	1083	1003	1058
2.45 p.m.	0.01	25.0	35.2	23.5	11.77	1835	3.2	0.08	1070	995	1050
3.00 p.m.	0.01	25.0	35.4	23.5	11.9	1855	3.4	0.08	1096	1005	1060
3.10 p.m.	0.01	25.0	35.4	23.5	11.9	1835	3.2	0 08	1083	998	1053
6-16-11											
10.00-10.15 a.m.	0.02	24.5	34.0	23.1	10.91	7425	11.3	0.3	1072	992	1048
11.30 a.m.	0.02	24.5	34.0	23.1	11.37	2375	3.6	0.1	1072	995	1047
11.00 a.m.	0.02	25.0	34.4	23.1	10.76	5040	7.5	0.2	1077	997	1050
11.30 a.m.	0.02	25.0	33 8	23.0	10.82	5050	7.8	0.2	1085	1002	1055
1.00 p.m.	0.02	25 0	33.8	23.0	9.6	5600	7.5	0.2	1068	988	1041
1.30 p.m.	0.02	25.0	32.5	22.9	9.48	5730	7.5	0.2	1080	1000	1053
2.00 p.m.	0.02	25.0	32.6	22.9	9.58	5720	7.8	0.2	1088	1005	1059
2.30 p.m.	0.02	26.0	32.7	22.9	9.78	5610	7.9	0.2	1090	1006	1060
4.30 p.m.	0.03	26.0	32.4	22.8	9.56	5730	7.6	0.2	1088	1007	1065
4.45-4.55 p.m.	0.03	26.0	32.4	22.8	9.65	11350	15.6	0.4	1087	1004	1062
		22.8	32.5								
6-17-11											
9.50 a.m.	0.02	23.0	31.3	22.3	9.0	6195	8.8	0.2	1107	1013	1061
10.15 a.m.	0.02	23.0	31.3	22.3	9.0	6190	8.4	0.2	1105	1015	1062
10.45 a.m.	0.02	23.0	31.3	22.3	9.0	6190	8.4	0.2	1105	1015	1062
11.10-11.20 a.m.	0.02	23.0	31.4	22.3	9.1	9175	12.4	0.3	1104	1016	1064
11.30 a.m.	0.02	23 0	31.6	22 3	11.34	4900	8.0	0.2	1103	1018	1066

$$\frac{0.00397\,AB}{D} = E; \quad E - \frac{2\ 13\,C}{D} = F; \quad \frac{30 \times (J + 273)F}{(\text{Barometer} + H) \times 290} = G$$

ELECTRICAL MEASUREMENTS

53 The output of the unit was metered with a shunt type watt-hour meter. This instrument was installed after the three preliminary runs were made, as

TABLE 8 EXHAUST GAS ANALYSES

Date	Sample	Time	Load	Engine	CO_2,%	O_2,%	CO,%
6-12	Snap	1.45 p.m.	¾	Left-hand	9.7	2.3	0.4
6-12	Average	2.00-3.00 p.m.	¾	Left-hand	8.9	2.7	...
6-12	Average	3.15-4.15 p.m.	¾	Left-hand	10.0	2.2	0.4
6-12	Average	4.15-5.00 p.m.	¾	Left hand	10.1	2.1	0.5
6-13	Snap	8.00 a.m.	Full	Right-hand	10.6	1.2	0.1
6-13	Average	8.00- 9.00 a.m.	Full	Right-hand	10.8	1.1	0.2
6-13	Average	9.00-10.00 a.m.	Full	Right-hand	11.2	0.5	0.3
6-13	Average	10.00-11.00 a.m.	Full	Right-hand	10.7	0.8	...
6-13	Average	11.00-12.00 a.m.	Full	Right-hand	10.8	0.6	0.3
6-13	Average	12.00 m.-1.00 p.m.*	Full	Left-hand	9.8	2.2	0.5
6-13	Average	1.00-2.00 p.m.	Full	Left-hand	10.5	1.7	0.3
6-13	Average	2.00-3.00 p.m.	Full	Left-hand	9.9	2.6	0.4
6-13	Average	3.10-4.10 p.m.	Full	Right-hand	10.5	0.7	0.4
6-13	Average	4.20-5.00 p.m.	Full	Left-hand	10.4	1.8	0.5
6-13	Snap	5.20 p.m †	Full	Left-hand	10.2	1.8	0.8
6-14	Snap	8.35 a.m.	Full	Left-hand	10.4	1.7	0.1
6-14	Snap	9.00 a.m.	Full		9.4	3.2	...
6-14	Average	9.00-10.00 a.m.‡	Full		9.0	4.0	0.3
6-14	Average	10.05-11.05 a.m.	Full	Right-hand	10.2	1.6	0.1
6-14	Average	11.20 a.m.-12.20 p.m.⊙	Full	Left-hand	9.1	3.9	0.2
6-14	Snap	12.00 m.	½	Right-hand	10.1	2.7	0.1
6-14	Average	12.30-1.30 p.m.	½	Right-hand	10.1	2.7	...
6-14	Average	1.35-2.35 p.m.	½	Left-hand	9.7	2.0	0.5
6-14	Snap	2.00 p.m.	½	Right-hand	10.1	2.1	0.1
6-14	Average	3.00-4.00 p.m.	¾	Left-hand	9.5	2.2	0.1
6-14	Snap	3.30 p.m.	¾	Right-hand	10.7	1.5	...
6-14	Average	4-05-5.05 p.m.	¾	Right-hand	10.4	2.1	0.2
6-14	Snap	5.30 p.m.	¾	Left-hand	10.4	2.1	0.2
6-15	Snap	8.00 a.m.	Full	Right-hand	10.3	0.8	0.1
6-15	Average	8.00-9.00 a.m	Full	Right-hand	10.8	0.5	0.2
6-15	Average	9.00-10.00 a.m.	Full	Left-hand	9.7	2.2	0.4
6-15	Average	10.05-11.05 a.m.	Full	Right-hand	10.5	0.4	0.4
6-15	Average	11.15 a.m.-12.15 p.m.	Full	Left-hand	9.4	1.6	0.5
6-15	Average	12.30-1.30 p.m	Full	Right-hand	11.0	0.6	0.1
6-15	Average	1.30-2.30 p.m.	Full	Left-hand	9.8	2.2	0.7
6-15	Average	2.35-3.35 p.m.	Full	Right-hand	10.7	0.7	0.4
6-15	Average	3.50-4.50 p.m.	Full	Left-hand	9.8	1.9	0.5
6-16	Snap	9.30 a.m.	10% overload	Right-hand	10.5	0.8	0.8
6-16	Average	9.15-10.15 a.m.	10% overload	Left-hand	10.0	1.3	0.6
6-16	Average	10-25-11.25 a.m.	10% overload	Right-hand	9.8	1.5	0.8
6-16	Average	11.30-12.30 p.m.	Varied, ½ and ¾	Left hand			
6-16	Average	1.15-2.15 p.m.	½	Left-hand	10.0	1.7	0.3
6-16	Average	2.15-3.15 p.m.	½	Right-hand	10.4	1.6	0.1
6-16	Average	3.15-4.45 p.m.	¾	Left-hand	10.1	2.0	0.1
6-16	Snap	5.30 p.m.	¾	Right-hand	10.8	1.6	0.1
6-17	Average	9.00-10.00 a.m	Full	Left-hand	10.0	1.3	0.5
6-17	Average	10.00-11.00 a.m	Full	Left-hand	9.9	2.2	0.4

* Gas hand regulating valves adjusted.
† Ignition changed 5 deg. at 5.00 p.m.
‡ Left-hand engine not firing evenly.
⊙ Load reduced one-half at 12.30.

it was found that the load fluctuations were too great to enable averaging the load with the indicating precision instruments with any degree of definiteness. This instrument was placed far enough away from the bus-bars to insure its not being disturbed by stray currents. Its surrounding temperature was also observed and together with the shunt, was calibrated accordingly. The calibration tests of the meter and the shunt resistance were made by the Westinghouse Electric & Manufacturing Company. The meter was found to be running about 2½ per cent slow. Accurate calibration curves were used.

54 The extra load to be made up over and above that from the shops to carry whatever percentage of rating was sought, was taken up with a water rheostat especially built for the tests. It consisted of two sheet-iron plates immersed in salt water and tied in with the main bus-bars with heavy cables. The depth of immersion of one of the plates could be regulated from a hand-wheel at the switchboard where the operator could observe the load on the indicating instruments.

TABLE 9 SPECIFIC GRAVITY DETERMINATION OF NATURAL GAS (W. VA.) AT AMERICAN LOCOMOTIVE COMPANY, ALLEGHENY, PA.

Date	Air		Gas		
	Time of Fall	Temperature	Time of Fall	Temperature	Specific Gravity at 62 deg. fahr., 30 in. mercury
6-16-11, 10.30 a.m. Official Record	02 : 19.7	81	01 : 57.9	74	
	02 : 20.7	81	01 : 57 7	74	
	02 : 20.0	81	01 : 57.4	74	
	02 : 19.8	81	01 : 57.6	74	
Average.....	02 20.0	81	01 : 57.6	74	0.692
6-17-11, 10 a.m... Supplementary Observations	2 : 49.3	70	2 : 36.8	70	
	3 : 00.0	70	2 : 47.5	70	
	2 : 54.7	69.5	2 · 11.3	70	
	2 : 49.0	69.5	2 : 32.0	70	
	3 : 73.0	69	2 : 30.7	70	
	3 : 05.0	70	2 : 32.0	70	
	2 : 56.8	69.7	2 : 31.7	70	0.736

Method of calculation: Air = $140 \times \frac{541}{523} = 145$ Sec. corrected to 62 deg. fahr.; Gas = $117.6 \times \frac{534}{523} =$ 120.5 seconds corrected to 62 deg. fahr.; Specific Gravity = $\frac{(120.5)^2}{(145)^2} = 0.692$.

55 The Sangamo meter was read at regular intervals during each run to check the steadiness of the load maintained on the unit.

56 The temperatures of the generator were observed, and the rise was abnormal. This has the effect of lowering the efficiency of the generator, to what degree, however, is not definitely known, but no allowances were made.

WATER

57 Jacket and piston water was measured by two Worthington disk meters connected in parallel for runs Nos. 1 and 2, but one of them was cut out for

the later tests These meters were calibrated at the Pittsburgh Water Department and the one used in runs No. 3 to 13, was correct, the other registering about 1½ per cent slow.

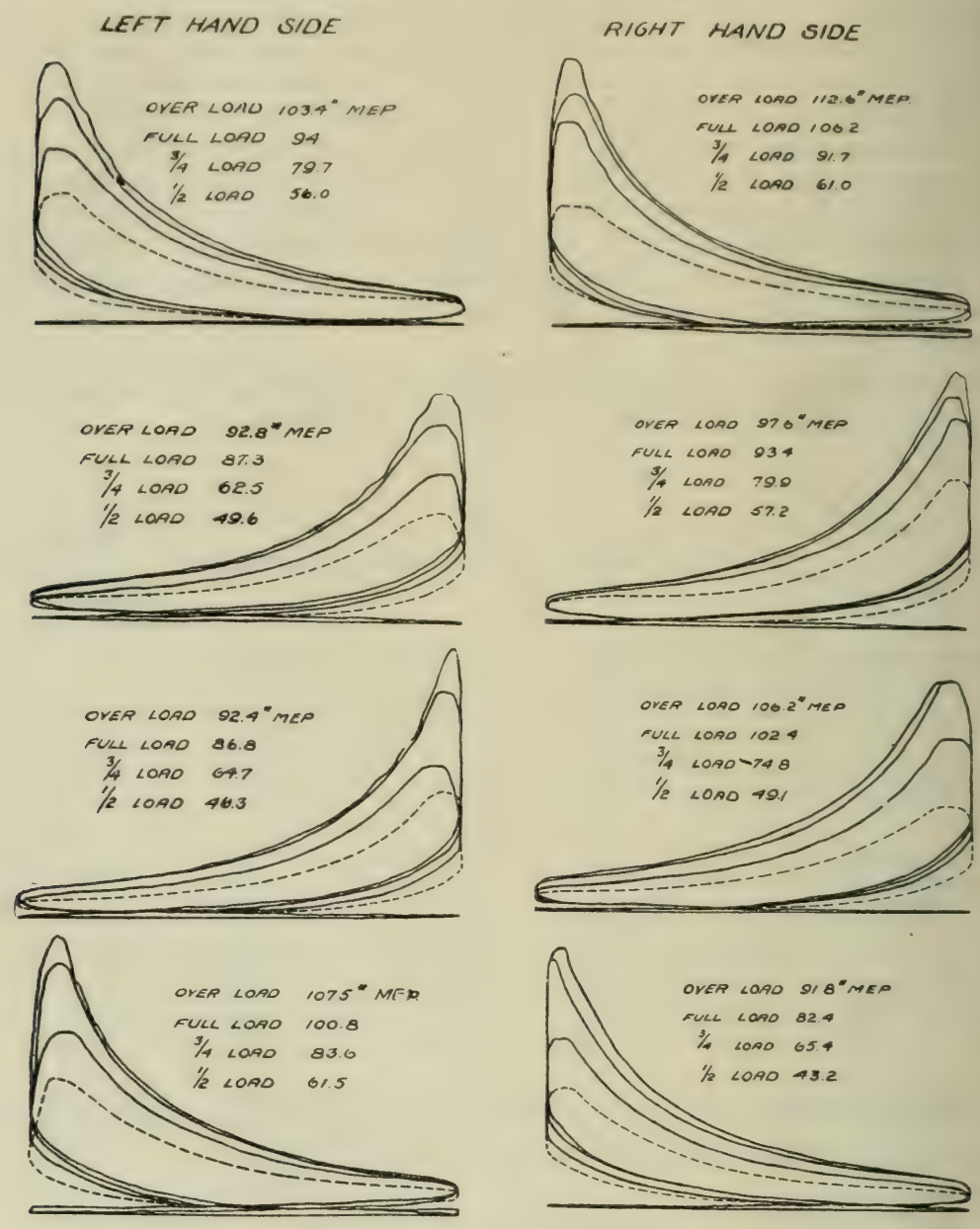


FIG. 8. TYPICAL CARDS TAKEN DURING TEST

58 The temperature of the incoming water was taken just after it passed through the meter. For the jacket temperatures, individual angle thermometers were placed in the outlet for each cylinder and an average taken. Angle thermometers were also placed in the drain from the tail rods for the temperature of the piston cooling water for each engine.

59 The amount of cooling water specified, about 7 gal per b.h.p-hr., depends upon an inlet temperature of 70 deg. For every rise of 5 deg. above this, the amount of water required is increased about 8 per cent.

60 At light loads the same quantities were allowed to flow through the engine, for which reason the rate per b.h.p-hr. became greater. If the water had been throttled, the consumption of water per unit output at the light loads would have been thereby reduced.

61 The average temperature of the incoming water was from 72 deg. to 75 deg. fahr., causing the water consumption to be a little higher. On the lighter loads this is probably due to a failure to cut the water down in proportion to the load.

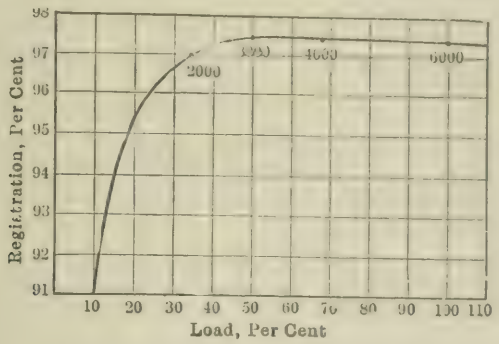


FIG. 9 CALIBRATION CURVE, WATT-HOUR METER

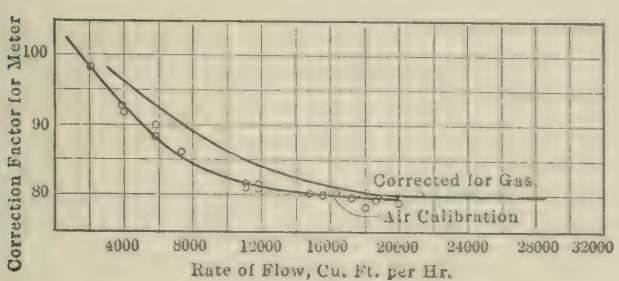


FIG. 10 GAS METER CALIBRATION CURVES

INDICATED HORSEPOWER

62 The engine was indicated primarily to discover if there were any conditions in any cylinder that might cause inefficient combustion. Practically no adjustments were made, and the engine was tested in the condition in which it was found. Cards were taken periodically to enable calculating the approximate mechanical efficiency (Fig. 8 and Table 3).

63 As the cards were used primarily to explore the firing conditions of the engine, their calibrations were not determined with any degree of precision. In fact, they are liable to be below normal scale, and therefore indicate too high mean effective pressure, this deduction being derived from comparison with similar tests.

OPERATING WITH ONE SIDE OF ENGINE DISCONNECTED

64 Tests were made operating one side of the engine by removing the connection on the opposite frame, to determine whether with long periods of light

demand on the engine the fuel economy would be sufficiently improved to warrant adopting this method in regular operation. Tests 11 and 12 were thus made with the left-hand engine, showing an improvement of about 15 per cent at half-load.

ACKNOWLEDGMENT

65 The authors wish to express their appreciation of the courtesy of the American Locomotive Company in permitting the publication of this informa-

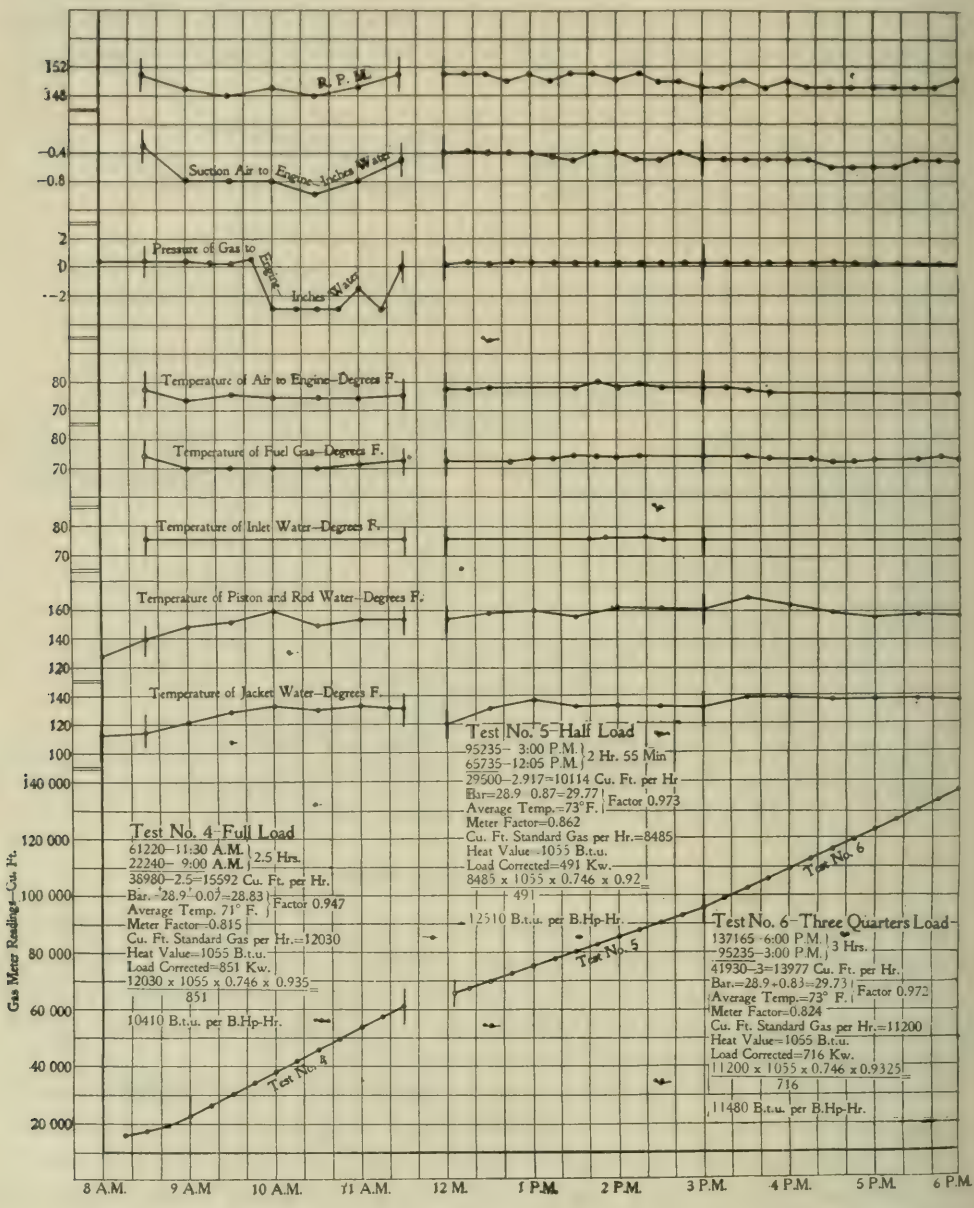


FIG. 11 SPECIMEN TEST LOG AND COMPUTATIONS

tion, and to announce the credit due the operating force, engineers from the engine builders' works and students of the Carnegie Technical Schools who rendered valuable assistance in carrying out the preceding performance tests.

DISCUSSION

W. D. Ennis. I happen to be able to give a little of the history of the engine in this plant, dating back to about 4 years before it was constructed. About 5 years ago the proposition submitted was that the engine was to be designed to be used on producer gas in the event of failure of natural gas. I notice in Par. 38 that gas delivered to the engine at 14 cents is equivalent in cost to coal of 12,500 B.t.u. at $3.50 per short ton. At that time the price of coal was something under $1.50 so that the advantage was entirely in favor not of building a gas engine to be used later for producer gas, nor a natural gas engine, but to be used at once for producer gas, which was the cheaper fuel with the prevailing price of coal. The only thing that kept us from doing it was that we were not sure that coal could be gasified and thought we would wait a few years to see. Meanwhile we contracted for the purchase of power at a couple of cents per kilowatt. The next step was to put in natural gas engines.

E. S. McClelland (written). I have been particularly interested in the value upon which, in Par. 40, the authors have determined for repairs and maintenance. A case has just come to my attention in which the record of repair parts furnished was desired. Two single engines of the same cylinder dimensions as referred to in the paper were installed in New York State during the latter part of 1906, and operated on producer gas, which contains foreign substances, such as dust and grit not present in natural gas. The total billing for repairs for the two engines for 5 years amounted to something under $2000. Adding 5 per cent to cover freight and labor charges, the cost for the 5 years may be conservatively expressed as $3000, or an average of $6000 per annum, or $1.66 per day. When it is considered that these engines are in constant use throughout the year in the exacting service of central station requirements, allowing very little time for the really necessary adjustments for proper upkeep, I feel justified in stating that the authors have not underestimated the repair account, but on the contrary, have allowed themselves ample margin for any possible contingency. This case is cited simply because I have investigated the actual facts, and have indisputable

evidence that plants of larger capacity have shown a very much lower repair cost under equally exacting service.

H. J. Freyn. In regard to the cost of installation of gas power plants, one of the main points at issue in the existing rivalry between steam turbine and gas engine plants, it should be pointed out that a number of larger blast-furnace gas-electric power stations in this country, including everything necessary for their operation was installed for less than $100 per kw., and that with present day experience in the way of building efficient gas-cleaning plants at reduced cost and judicious saving all along the line, this total cost could in the future be easily maintained and even reduced to about $95 per kw.

John Willis[1] (written). We have in our power house at James City, Pa., five natural gas engines of practically the same design as those mentioned in this paper, all direct-connected to alternating-

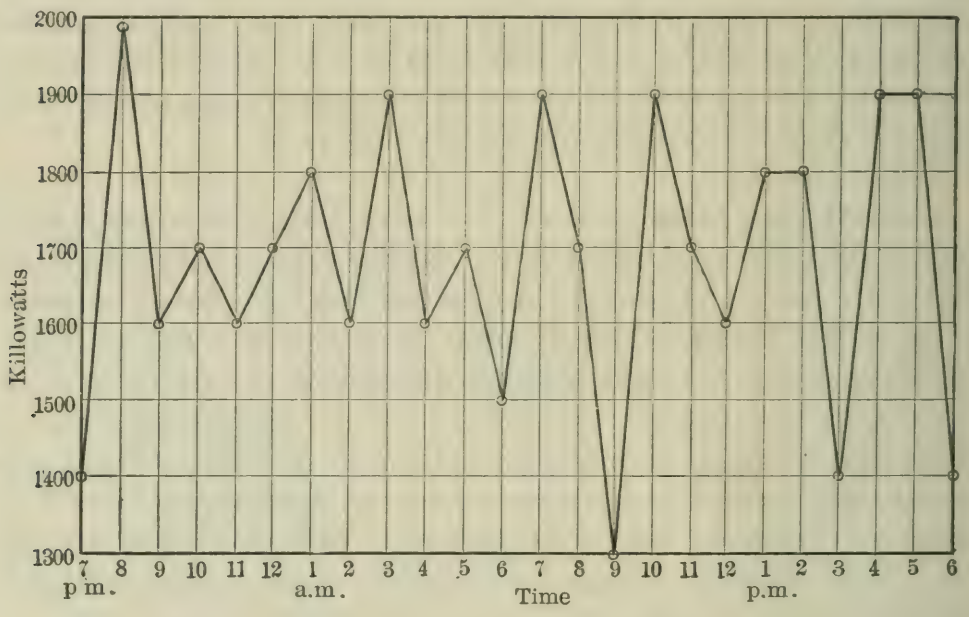


Fig. 12 Various Grades During 23-Hr. Run

current generators, three to 400-kw. generators turning 180 r.p.m. and two to 625-kw. generators turning 150 r.p.m., making 2450 kw. rated capacity. These operate on natural gas and the plant is used for furnishing power for the manufacture of plate glass. Our load is very unsteady, running from 1000 to 2000 kw. inside of 5 minutes. Our daily output is about 42,000 kw-hr., and our gas consumption

[1] Secy., Amer. Plate Glass Co., Kane, Pa.

is about 810,000 cu. ft. in 24 hours. For the month of June 1911, which may be taken as a fair average of what our power plant is able to accomplish, we operated 576 hours, with an output of 877,500 kw-hr. and the cost of operation, which was figured very accurately, was:

	Per Kw-Hr.
Fuel	0.22800
Labor	0.09800
Oil and grease	0.03000
Supplies	0.00051
Water	0.01400
Repairs	0.00400
	0.37451

Our plant is located 2000 ft. above sea level, and in explanation of the fluctuation of the load Fig. 12 shows in detail the various grades during the 23-hour run.

We are realizing very good economies and are more than pleased with the performance of the engines and with the satisfactory results obtained from the installation.

Howell C. Cooper[1] (written). To take up only one of many interesting points brought out by this test, is there not some better and more accurate method of determining the volume of fuel gas supplied to engines when under test than by the use of the instruments and means now generally employed by engineers for this purpose?

The method described by Messrs. Dreyfus and Hultquist consists in using one of the standard makes of gas meters and correcting the readings shown by it after finding its error by calibration. Since a mechanical meter, even though it be tested and adjusted before being placed in use cannot be relied upon to give correct results, the accuracy of the final determinations depends therefore entirely upon the accuracy of the calibration tests.

Calibration tests must therefore be made at the time the engine is being tested, for the reason that meter adjustments and actions are too apt to change in short periods of time, and it is the exact error of the meter reading at the time of the engine test which is being sought. And further since it is a well-known fact that transporting a meter from place to place is apt to cause changes in its adjustment, therefore moving it from its place of use to some other place for calibration throws doubt upon the accuracy of the results.

[1] M. E., Hope Natural Gas Co., and Peoples Natural Gas Co., Pittsburgh.

About the only way to calibrate the meter at the time of the engine test, and without disturbing or moving it, is by means of an orifice prover as described in this report, and this is indeed a very difficult instrument to use correctly and from which to get accurate results, especially when operated in the open air, as is necessary with gas. Many readings and much time are also required in order to obtain reliable results and to eradicate errors introduced in the readings, and the changing conditions of back pressure due to wind, rain, or drafts. It is not always convenient or safe to blow quantities of gas out in confined spaces where the fuel meter may be located. For all these

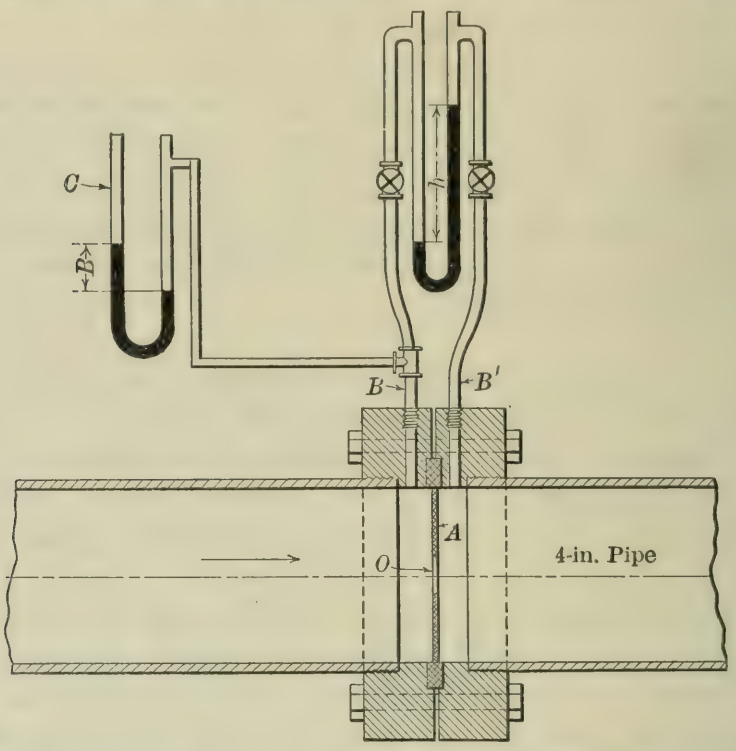


FIG. 13 STANDARDIZED ORIFICE HELD BETWEEN TWO FLANGES IN FUEL SUPPLY LINE

reasons it seems to the writer that some better method and form of measuring instrument is urgently needed by engineers for use in this important matter of gas-engine testing.

Two methods suggest themselves for consideration. One is the electric meter which weighs the gas by means of measuring the quantity of electrical energy required to heat the gas a known and constant amount and thereby to calculate the volume or weight of gas passing through the meter. Commercial forms of these meters have been brought out and might be modified to some special form for the special use of gas-engine testing. The other method and the one

which the writer feels is preferable, is the use of a standardized orifice cut in a thin plate and held between two flanges placed directly in the fuel supply line. Fig. 13 shows an orifice arranged in such a way. A is the plate in which the orifice O is cut; B and B' are the pressure holes by which a water U-glass is connected across the plate A for measuring the differential pressure h which causes the flow through orifice O. The water or mercury column C, is also connected to the up-stream side of the orifice to obtain the static pressure P above or below the barometer.

The laws governing the flow of gas through an orifice arranged as shown in Fig. 13 are the same as for a pitot tube, but as it is not practical to construct pitot tubes of the small diameters which would be needed for measuring the volumes of gas used for fuel on gas engines, it is here proposed to use the orifice. In constructing this instrument it is well to choose a 4-in. diameter pipe as this permits the use of several sizes of orifices. A standard design of flange-union can be made so that the pressure openings shall be a fixed distance from the respective faces of the orifice plates. The orifice plates are each placed in this flange-union, connected up and a series of readings of different combinations of P and h made. The gas passed for each reading is measured either in a holder, by means of another orifice already calibrated, or by an orifice prover under favorable conditions. In this way a coefficient C is calculated, which is the amount of gas Q which will pass through the orifice, in some chosen period of time, for a value of h equal to 1 in. of water and a value of P equal to 1 lb. absolute pressure. Q is calculated to give the quantity of gas in some fixed condition of pressure temperature and specific gravity. Then for any observed value of P and h we have

$$Q = C \times \sqrt{h\,(P + \text{barometer})}$$

This quantity is corrected for temperature and specific gravity should they be observed to differ from those conditions on which Q is based. By calibrating a number of different sizes of orifices in a standard flange-union, they may be used in another flange-union which is made exactly like the standard.

It must be understood that an orifice placed in a run of pipe in the manner shown cannot be used as a measuring instrument except for a limited range of velocity or value of h, for the velocity of approach becomes a factor and the coefficient would have to be modified. But the writer has found by experiments and many practical tests that a steady consistent coefficient can be established for orifices used in

this manner for velocities up to 30 in. of water reading h. This is a liberal range for practical use as it is easy to choose a suitable size of orifice to give readings within this range.

The limits of this discussion will not permit of a mathematical deduction of the equation, but it is well known from use with pitot tubes. The writer believes that those engineers who have to make many gas-engine tests would welcome a standard and accurate method of getting the fuel consumption of gas engines, and deems it a matter of sufficient importance to merit the attention of the Society to the end that some standard method and instrument can be devised and supplied to those who wish them for purposes of gas-engine testing or any similar use. Results thus obtained would be approved.

THE AUTHORS. Professor Ennis has made note of some interesting facts in regard to the installation of this unit. It might be stated that this engine has been arranged with suitable piping and connections so it may be easily changed over for producer gas operation. In reference to the point of the equivalency of the existing price of gas to $3.50 coal, and the fact that coal in the Pittsburgh locality is less than $1.50 per ton, these deserve some additional explanation.

On the B.t.u. basis alone, 14 cent gas (1000 B.t.u.) represents the same cost per 1,000,000 B.t.u. in the fuel as would be the case of coal of 12,500 B.t.u. per lb. at $3.50 per ton. This comparison obviously shows an advantage in favor of the coal as the cost to gasify has not been included. When fixed charges are also taken into account, the relationship is materially affected, and to illustrate, Fig. 14 has been prepared. It will be seen that at 100 per cent load factor, 20-cent gas equals coal at $1.50 per ton, or for 50 per cent loading, 25-cent gas will equal the $1.50 coal. The reasons for the selection of this type of unit are hereby manifest.

It has been very gratifying to have had the items of cost confirmed in a large degree by the experience of others. There is one item in the table of construction costs which might be qualified, and that is the engineering expense. This being a simple layout and, moreover, one for an industrial plant, there was no occasion for organizing and maintaining a special engineering force, hence negligible cost of drawings and subdivision of supervision in other parts of the plant, which reduces this item, therefore, to a relatively small percentage of the entire cost.

Mr. Cooper has taken up some important features regarding meters for testing purposes. It is plain his recommendations are of much

value for this work. The feeling of the necessity for calibrating the proportional meter with extreme care, is shared with Mr. Cooper, and accordingly every precaution was observed on these tests to satisfy any criticisms which would naturally suggest themselves when the meter is used in the ordinary way. It is true that there are, in many cases, objections to placing the orifice prover out of doors, due to the disturbing influence of air currents, but in the above instance, the

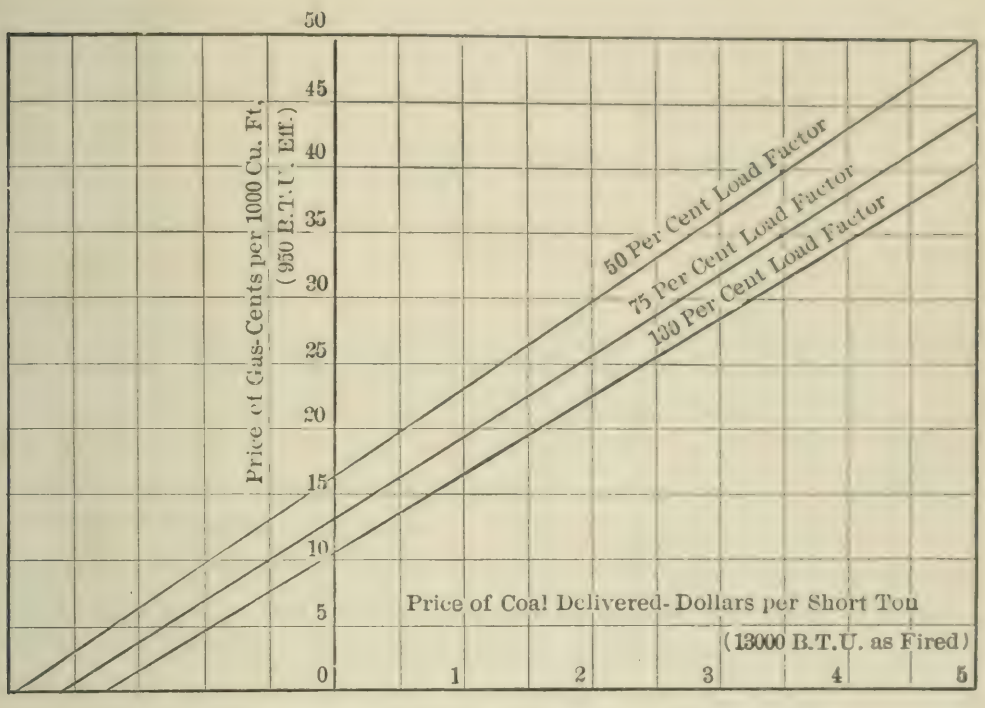


FIG. 14 EQUALIZATION CHART, COAL AND NATURAL GAS

meter was protected not only by the side of the building, but tests took place during hot, sultry days when virtually no breeze was astir. More extensive use of Mr. Cooper's meter is to be encouraged, and it is further urged that it be equipped with an accurate recording device. The proportional meter, being immediately available, naturally received preference in our tests, and being provided with a register, was of great value in integrating the consumption. It was inspected and tested in all details.

No. 1339

THE DEVELOPMENT OF THE TEXTILE INDUSTRIES OF THE UNITED STATES

A GENERAL STATEMENT OF PRESENT CONDITIONS

By Frank W. Reynolds,[1] Boston, Mass.

Non-Member

The importance of the textile industry in a country which, like the United States, now manufactures all the woven fabrics really necessary for the clothing of all its people, hardly needs to be argued. Yet in the case of our own country progress in the textile industry has been so rapid, and its development has had such a vital relation to other industries, that some statistical evidence of its great place may serve as an inspiration to the men, mainly the mechanical engineers, who are responsible for its continued progress and increasing efficiency. Owing to the fact that the figures from the census of 1910 are not yet available as a complete survey of the textile industries, it is necessary to cite here the figures of the census of 1900; but it should be noted that these figures do not adequately suggest the great expansion in the textile industries that has taken place since the enactment of the tariff law of 1897. Some later figures are indeed available, but even with the old figures it is possible to gage quite accurately the present position of textile manufacturers. The statistics which follow cover only the cotton, the woolen, and the silk industries, in what the census reports term the "proper" sense of those words, namely, the production of fabrics either ready to wear, or ready for the dyeing and finishing processes which precede the conversion of certain cloths into made-up clothing.

[1] Member of firm, Lockwood, Greene & Co.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

TABLE 1 VALUE OF DOMESTIC PRODUCTS, EXPORTS, IMPORTS FOR CONSUMPTION, AND TOTAL CONSUMPTION OF TEXTILES, WITH PER CENT OF IMPORTS TO TOTAL CONSUMPTION, FOR THE UNITED STATES, 1900

	Value of Domestic Products	Exports	Domestic Consumption	Imports for Consumption	Total Consumption	Per Cent of Imports to Total Consumption
Total	$743,447,062	$25,556,057	$717,891,005	$82,214,010	$800,105,015	10.3
Cotton manufactures	339,200,320	24,003,087	315,197,233	39,789,989	354,987,222	11.2
Wool manufactures	296,990,484	1,300,362	295,690,122	15,620,487	311,310,609	5.0
Silk manufactures	107,256,258	252,608	107,003,650	26,803,534	133,807,184	20.0

TABLE 2 COTTON SPINDLES IN THE WORLD IN THOUSANDS, 1900

Great Britain	46,000
Continent of Europe	33,000
United States	19,008
India	4,400
Japan	1,500
China	600
Canada	640
Mexico	460

TABLE 3 VALUE OF SILK PRODUCTS OF EUROPE AND THE UNITED STATES, 1900

Countries	Value of Products	Per Cent of Products
Total	$395,000,000	100.0
France	122,000,000	30.9
United States	92,000,000	23.3
Germany	73,000,000	18.5
Switzerland	38,000,000	9.6
Russia (in Europe)	21,000,000	5.3
Austria	17,000,000	4.3
Great Britain	15,000,000	3.8
Italy	13,000,000	3.3
Spain and Portugal	4,000,000	1.0

TABLE 4 COMPARATIVE SUMMARY, BY INDUSTRIES FOR THE UNITED STATES

Industries	Year	Number of Establishments	Capital	Wage-Earners		Cost of Materials Used	Value of Products
				Average Number	Total Wages		
Combined Textiles	1900	4312	$1,042,997,577	661,451	$209,022,447	$521,345,200	$931,494,566
Cotton manufacture							
Cotton goods	1900	973	460,842,772	297,929	85,126,310	173,441,390	332,806,156
Cotton small wares	1900	82	6,397,385	4 932	1,563,442	3,110,137	6,394,164
Wool manufacture	1900	1414	310,179,749	159,108	57,933,817	181,159,127	296,990,484
Silk manufacture	1900	483	81,082,201	65,416	20,982,194	62,406,665	107,256,258
Hosiery and knit goods	1900	921	81,860,604	83,387	24,358,627	51,071,859	95,482,566
Flax, hemp and jute	1900	141	41,991,762	20,903	6,331,741	32.197,885	47,601,607
Dyeing and finishing textiles	1900	298	60,643,104	29,776	12,726,316	17,958,137	44,963,331
Combined textiles	1890	4276	767,705,310	517,237	168,488,982	$447,546,540	759,262,283

TABLE 5 COMPARATIVE SUMMARY, NOT INCLUDING FLAX, HEMP AND JUTE, WITH PER CENT OF INCREASE FOR THE UNITED STATES, 1900

		Per Cent of Increase
Number of establishments	4,171	1.4
Capital	$1,001,005,815	35.3
Wage earners, average number	640,548	27.7
Total wages	202,690,706	23.9
Cost of materials used	489,147,315	16.1
Value of products	883,892,959	22.4

2 Even the statistics of a decade ago are impressive evidence of the enormous strides made by the textile industries in this country during the last century. At the beginning of the nineteenth century the textile industries were represented by spinning mills for cotton yarn, by wool-carding factories, and fulling mills, in which cloth woven on hand looms in the households of the country was dressed and prepared for sale. It was not until 1814, less than a hundred years ago, that the first broad power loom for cotton goods was worked out by Francis C. Lowell, of Boston, and put in operation at the new plant at Waltham. From that time, progress was rapid, and in the cotton industry particularly, almost numberless improvements in machinery were made during the next decade. The actual complete establishment of the cotton industry in this country as a factory industry depending wholly upon power-driven machinery dates from the establishment of the Middlesex Mills at Lowell in 1823.

It was a year or two after this that the first broad power looms were introduced into wool manufacture. The history of our textile development since that time is too intricate to be summarized in this paper. It must be sufficient to note that now the United States is the second greatest producer in the world, not only of cotton and woolen cloths but of silk; while the use of linen for carpet yarns and thread, and for towels and towelling has reached large proportions. Besides the textile industries "proper," there are also a great multitude of small factories engaged in the production of textile specialties.

PRESENT CONDITIONS AND PROBLEMS

3 The textile manufacture, probably to a greater extent than any other industry in the country, has tended to a complexity, both mechanical and operating, that is rather exceptional. This has been the result of two main factors: the peculiar conditions imposed by the character of our most used natural fiber, cotton; and the constant necessity for a progressive decrease in the labor expense of cotton manufacture. The cotton industry has been shaped, as no other industry has, by the limitations found in the raw material itself. The necessity for humid air in cotton factories, both to affect the physical condition of the fiber, and to prevent the mischief caused by electricity in the fibers during manufacture, is a requirement of great importance, and one that has not even yet been satisfactorily met. The character of the cotton fiber also, whether well or ill moistened, imposes certain limitations upon the speed of the machinery handling it. On this side, the speed at which roving is now handled seems to have reached the practicable maximum, and the same thing is true of looms. In the spinning frame, however, it has been shown that the cotton fiber will stand a more rapid handling than it now receives.

4 In addition to these physical limitations of the cotton fiber itself, the cotton industry has had to face another rather peculiar series of problems in the cost of labor. The automatic machinery of today represents the constant effort to cut down the labor cost of production, this labor cost bearing a rather higher proportion to the total production cost in the cotton industry than in most others. Another labor problem has arisen through the changes that have taken place in the available labor supply. The disappearance of the native American labor of the middle of the last century has been followed by a greatly varied succession of labor

elements, until the industry in many places depends upon a class of alien operatives whose habits and standards are not by any means perfectly adapted to the most profitable manufacture. These latter problems, of course, emphasize the desirability of more completely automatic machinery, while they also impose new requirements in factory organization and administration.

5 The textile establishments of today, owing to the intricacy and importance of conditions, some of which have just been noted, are making full use of the best engineering skill in the design, construction, arrangement and equipment of plants. Many well informed and practical men whose work has been wholly devoted to the textile manufacture, have studied with great care the problems involved, and have made the solving of them their whole aim. The result is a highly specialized work, whose principles have become well understood and applied in all of the structural and mechanical features of the later new plants, and in the reconstruction of the older plants. Advantage has been taken of all new forms of construction wherever practicable, improved machinery for manufacturing and for power purposes has been installed to the fullest extent, and in many instances the initial trials of new forms of mechanical and electrical improvements have been made in textile mills. We find in these modern plants, buildings of either slow-burning construction or of reinforced concrete, equipped with the most modern systems of power transmission, of fire protection, electric lighting, humidifying apparatus, and the latest type of manufacturing machinery of both domestic and foreign design.

6 In spite of the great accomplishment already achieved, however, there still remain unsolved problems, and other problems of many kinds in which the cotton industry has so far only approximately reached a satisfactory position. These problems are in the main such as it is the proper function of the mechanical engineer to solve, and therefore today, as in the early period of the cotton industry, the mill calls upon the machine designer and the machine shop for better tools and better methods.

7 It is to be borne in mind that textile mills in common with other industrial plants are built for a specific purpose, which is to produce an income for the owners. All engineering problems in connection with them must be subject to this condition. It would seem very impractical, to say the least, to erect a building or to purchase machinery for that building which would be so expensive that the profit would be diminished because of the interest on unneces-

sary expenditure. The managers and designers of textile plants of all kinds must bear in mind that the productive power of machinery equipment, whether for the manufacturing process, for power, light, heat or other departments must not be so great that a proper return upon the investment is not possible.

8 It will be found upon investigation that the power plants of textile mills are as a rule especially economical and that advantage has been taken of nearly every labor-saving device and of the most approved machinery which can be obtained. At the same time managers of these plants have endeavored not to spend money which will not show a reasonable return, and it should be the aim of all engineers who approach the problems to be solved in connection with this industry, that while the most efficient machines and processes are desired, the cost of the installations of machinery necessary to produce such results must be kept at a figure which is not prohibitive.

9 The arrangement of the buildings of the plant is one of the most important features to be considered. Much unnecessary labor is often expended because of the poor arrangement of buildings forming different parts of a group, and the equally poor arrangement of the machinery within the buildings. It must be borne in mind that the least number of motions and the shortest distances to be traveled play some of the most important parts in economy of production. Whether buildings should be one or more stories high or of a greater or less width will be determined by a consideration of these points. A brief survey of present conditions and needs may be helpful.

CONSTRUCTION

10 Nearly every engineer is familiar with what is known as the slow-burning type of mill construction. It was first brought out by some of the early designers of cotton mills, and has since been used in some form or other in nearly every other type of industrial plant. For many years this form of construction, with slight variations, has been successfully used and has been regarded as sufficient for the needs of the industry. It has developed however, that a better form of construction is available and to this form of construction attention is called in this paper.

11 A rigid, non-vibrating mill building is one of the immediately desirable things. The slow-burning mill construction has great virtues, but rigidity is not one of its possessions. Rigidity has been

found to produce marked economies in the maintenance of textile machinery, as well as machinery of many other types, and for this reason alone the modern type of reinforced concrete construction has proved itself highly desirable. The friction between concrete floors and machine bases is so great that with much fewer bolts than are required on a wooden floor, machinery can be so securely anchored that it will not walk out of its proper position. In consequence, shafting and machines keep in better alignment, there is less wear and tear on the machines themselves, and the cost and bother of repair are greatly diminished. Simple and effective ways of attaching shafting and motors to concrete beams have been devised, so that from the point of view of machinery installation and maintenance the rigid-frame mill building of reinforced concrete has proved itself superior to any other type. The cost of reinforced concrete is somewhat greater than that of mill construction, as a general thing; but there often exist in some localities special conditions in which concrete is as cheap or cheaper than mill construction. Against its usually higher cost there may properly be set its indestructibility by fire. The concrete building not only does not burn, but if properly built, it is so little harmed even by a serious fire that it can be equipped with new machinery and put in operation again in a very small part of the time needed to replace a damaged mill-construction building. There is a species of fire loss, the interruption of production, for which insurance is often an inadequate compensation; the concrete mill building practically eliminates this loss.

12 Another valuable advantage of the concrete mill building is that it allows a much greater window space than is possible with anything else but steel construction, which latter, if effectively fireproofed, is much more costly than concrete. A concrete building can have as much as 80 per cent of its outer wall area devoted to lighting, as against a maximum of about 50 per cent in mill construction. Existing provisions for the escape of operatives in case of fire in cotton mills are on the whole exceedingly inadequate. Most mills are equipped, in addition to stairways with the familiar type of outside fire escape consisting of galleries or balconies for each story, with ladders from the top of the series to the ground. The actual operation of this type of escape is that the population of a big workroom is, in case of fire, crowded upon a series of outside balconies from which movement to the ground is excessively slow under the best of conditions; and in case of a fire panic, is but little less dangerous than the fire within the wall. If a fire occurs in a lower floor of

the mill, it breaks from the windows of that level and the ascending smoke and flames instantly render useless the whole tier of fire-escape balconies above. It would seem that nothing more faulty in principle, or imperfect in execution, could well be imagined, and it is a fair cause for astonishment that these contrivances should for many years have been accepted by engineers.

13 The proper and only adequate device in the way of a fire escape is the isolated staircase tower. It is doubtful if the building of such towers in adequate number would at all seriously increase the construction cost of cotton factories. It is evident that only the isolated stair tower can be a real fire escape, and it would seem to be equally evident that this type of safeguard should be provided for in every new mill, and that every effort should be made to provide for it in existing buildings. It may be objected that disastrous fires in textile mills are comparatively rare, and this is indeed true. This objection, however, is beside the point. Mills several stories high cannot be built utterly without fire escapes, and if the necessity of some form of fire escape is once admitted the argument for putting in a really efficient device is a strong one.

POWER

14 The changing conditions of power production and transmission have from the earliest days of the cotton industry had the most potent influence in shaping the material and conduct of the whole industry. The first effect was in forcing the location and even the shape of mill buildings to a large extent to suit the possibilities of the land lying in immediate contact with water privileges. As a result of building the old mills as close as possible to the water-wheel which gave them power, mills were made high and narrow and often built on curves which not a little embarrassed the mechanical transmission of power; and groups of buildings were strung out in a long line, or arranged in other peculiar groupings which the practice of today recognizes as unsuitable for effective power transmission, or for the careful arrangement of the sequence of processes, so important in the economy of the modern textile mill.

15 Aside from these direct effects on the type of textile mill construction, the power element has had other large effects which it is perhaps not easy to separate under distinct headings, though the results form together a series of complications which are familiar to every mill man. The changing conditions of power transmission

have had much to do with the grouping of machines of the same type within the mill, and even the machinery layout of the whole plant. The earlier systems also had much to do with accidents to operatives.

16 The modern use of electrical drive, either on the group system or by individual motors, dating from its first general employment in the Columbia Duck Mills in 1893, has had a revolutionary effect on many aspects of the cotton industry. It removed, at the outset, the former supposed necessity for building mills in immediate contact with water powers, and therefore permitted the arrangement of the different buildings of a plant to satisfy the requirements of speed and economy in the routing of material and product. With the modern electrical transmission line, mills may be built miles from the power generating station, in places where the supply of labor, of building materials, or of transportation facilities make it most advantageous to locate the plant. For situations of this general type, where power must be developed at water sites which do not offer the other conditions required for a mill, electric driving may hardly be disputed as the best form of power.

17 In other situations, however, particularly where power has to be obtained from steam, it is still a matter of debate among engineers as to whether electrical transmission or mechanical is the more efficient and economical. The writer believes that when power can be taken from the flywheel of the engine by a short, direct drive to the head shafts of the mill, such form of transmission is preferable to electrical drive. But whenever there is a scattered group of mills and especially where transmission shafting has to make angles, electrical transmission is decidedly the more desirable. The differences of opinion as to these two forms of power transmission are due in large part to the fact that, although the transmission losses are greater with electrical transmission than with mechanical, the motor-driven machine gains certain other things which make the equation more than the simple matter of comparing transmission losses; for example, the group system of motor drive in which losses are slightly greater than with mechanical transmission.

18 As usually arranged with mechanical drive, an average of about 72 or 74 per cent of the i.h.p. of the engine reaches the machine. With motors driving groups, about 65 to 70 per cent of the i.h.p. of the engine reaches the machine; while with the individual motor drive the similar efficiency will be from 70 to 75 per cent.

19 The virtue of electrical transmission which largely counterbalances its lower efficiency is its constant speed. The advantage

of the motor is most apparent in individually motor-driven machines, where speed is even because of the positive action of the motor. It is true that there is a drawback to the use of individual motors, and even of group driving motors, in the high cost of motors in comparison with shafting, but even with the losses of power already mentioned, the advantages of the present type of motors will in many cases outweigh this excess of cost. The relatively low efficiency of the small motor seems to be the chief weakness of the individual drive so far as regards efficiency of power transmission. The small motor seems to offer an excellent field for improvement at the hands of the engineer.

20 Shafting bearings that will largely reduce the friction losses, which are too great even with the modern ring-oiled babbitted bearings, deserve the earnest attention of mechanical engineers. The solution seems to lie either with roller or ball bearings, and it is possible that ball bearings, properly designed and constructed with extreme accuracy of measurement from the proper qualities of steel, may prove the best type. Roller bearings have in many cases proved very efficient but it would seem that the length of contact with the shafting which they involve makes them inherently more liable to derangement than are the ball bearings. Theoretically, at least, since a ball-bearing hanger involves only a single line of contact points, it would seem that a bearing of this type might be less liable to derangement, and would offer greater latitude of adjustment than a roller bearing.

21 Where the ordinary type of shafting bearing is retained it is often possible to reduce friction losses as well as lubrication cost, by a more careful study of lubrication. Experience has shown that the necessary study and care of the cost differences between the best and merely ordinary methods on the common types of lineshafting, are more than repaid by the results. It should be obvious, however, that merely as a mechanical question the roller or ball bearing is greatly superior to any sort of sleeve contact.

22 Two special problems under the head of power deserve more attention from engineers. The first of these problems, which is how coal or ashes can be handled more economically in a given situation, may seem a trifling matter; but the effect of bad arrangements in this particular has an easily visible effect on operating expense.

23 The other problem is provided by the power situation in many New England mills where the water power has proved insufficient for expanding plants which now depend upon a combination of steam

and water power in which the latter, once the principal source, has very often become the auxiliary. There is considerable opportunity for the engineer in the proper arrangement and economical design of such combination power plants.

MACHINERY

24 With regard to the machinery concerned in the manufacture of the cotton fiber, it is rather noteworthy that the producing capacity of 100,000 sq. ft. of mill floor has not been materially increased within a number of years. There has been a large development in making machines more automatic in their operation, with a consequent reduction in the labor cost, but increased rapidity in the processes of manufacture has been far less in evidence in the cotton than in many other industries. Apparently this situation is due to the fact that with the exception of spinning machines, cotton machinery of the existing types is already running at as high speeds as the nature of the fiber will allow. In the ring-spinning machine it seems fairly sure that when certain improved devices already in operation have been brought to perfection, spindles will run 15,000 r.p.m. where the maximum is now ordinarily about 9000 or 10,000. As the labor cost in the cotton industry has also been reduced less than in any other industry, it becomes a pertinent question for the mechanical engineer to consider whether the existing processes really represent substantially the maximum possible speed at which the fiber can be handled. The writer does not wish to assert that radically new types would make possible a great increase in speed, but with the facts before us it may be reasonable to suggest the desirability of making a careful study of the chances for improvement by a departure from present types of machinery. While the fact is not proof that much better types of machinery are still possible, it is true, nevertheless, that the existing methods for the manipulation of the cotton fiber are those of tradition. The question that might well be considered is whether the pioneers in cotton processes, on which succeeding generations have merely made piecemeal mechanical improvements, hit upon the only principles by which cotton can be successfully manufactured; or whether, on the contrary, more rapid production can be attained by the adoption of some different methods of manipulation. If the principles of the present processes are to be considered a finality, it would seem that the only possible gains will be small, and accomplished by making existing types of machinery more fully

automatic, and by training the necessary operatives to a condition of higher efficiency. There is some room for improvement in this latter direction when it is realized that the average producing output of textile machinery is about 85 per cent of the theoretical capacity of the machines, while some of the best mills have maintained an efficiency of well over 90 per cent.

25 There is a chance to make considerable saving by the combination of some of the processes which are now performed separately. As an example, it is possible to do away with the spooling machine, and a machine has already been designed for this purpose, although it is not in general use in this country. Careful study would probably show that this plan of concentration could be carried still further.

26 The providing of satisfactory machinery for the manipulation of cotton waste is another problem, the thorough solution of which would benefit both the machine manufacturer and the cotton mill.

HUMIDITY AND TEMPERATURE

27 One of the most important unsolved problems in the cotton manufacture is that of maintaining within mill buildings the temperature and humidity necessary for the most successful handling of the cotton fiber. The almost universal cooling off of the air of mills during the night and over holidays greatly reduces the absolute humidity in the cold season, even if it does not lower the relative humidity of the air. But a high relative humidity at a low temperature does not put the fiber in the best condition for working. There is needed not only a certain absolute humidity, but in addition a temperature which allows the fiber to absorb the moisture in a way to bring it to its best condition.

28 Most of the humidifiers now in use, those of the self-contained type, are essentially the same in principle as the primitive method of admitting steam through rose heads; they depend upon throwing a fine spray of vapor into the room, where it must be absorbed by the room air and diffused until it meets and affects the fiber under manipulation. The true principle would seem to require that heating and humidifying be combined, those two functions, and also that of ventilating, being performed by the same equipment. With the positive-pressure air system of heating, temperature can be regulated automatically and a uniform temperature can be maintained. It seems evident that the best system for humidifying will be that in

which the entering current of air for heating and ventilating is also provided with the requisite degree of humidity before it enters the room. It is certainly as possible to secure automatic control of humidity in the entering supply, as automatic control of temperature. Such a system, while it might be difficult to reduce it at the outset to perfectly reliable action, is in principle much simpler than any system of numerous heads distributed throughout a room; and under adequate study and experiment it will probably turn out to be simpler also as a piece of mechanism. By such a system it should be easy to secure not only proper conditioning of the fiber itself, but also proper conditioning of the operatives, the latter being quite as important in the long run as the state of the fiber.

29 The maintenance of a uniform temperature and humidity during all the non-working hours would greatly improve the condition of the fiber. When, as is often the case, mills are allowed to cool down during the night, either as a result of mistaken notions of economy, or through the carelessness of watchmen, the cotton cannot be in the best condition for manipulation at the beginning of the working hours. It must wait for its proper conditioning until the mill has slowly reached the temperature, and the air the proper humidity. There is also required in the heat of summer some adequate way of cooling the room air when the outside temperature is too high. Both results can best be obtained by combining the supply of moisture with the supply of hot or cool air as the case may be.

REMOVAL OF LINT

30 Another unsolved problem is some method of removing lint from machinery, motors, and wall, ceiling and belt surfaces that shall prevent soiled lint from damaging the quality and appearance of the product. In some mills this task is handled by having machines and surfaces continually wiped off with cloths, a method that is not entirely efficient, and undeniably costly. The practice of blowing lint from machines and surfaces with a blast of compressed air is familiar, and also unsatisfactory. The lint blown from the motors and from belts, walls and ceilings is in large part diffused again into the air, and tufts, often soiled, descend upon the cotton in process of manufacture and appear as more or less soiled lumps in the finished product. The air blast is also objectionable for the way in which it breaks or disturbs yarn or roving in the process of production. The lint evil is naturally most serious in the carding and weaving rooms.

For the former there is already a portable pneumatic suction machine which can be removed from card to card in order to take up the lint set free when each card is cleaned. The device is successful in removing a considerable part of the lint, but it does not cover the whole problem. The practical difficulty lies not so much with the system itself, for pneumatic suction is adequate, and we know how to use it, but in applying it without becoming involved in prohibitive expense. It is of course mechanically possible to provide a system of exhaust hoods, even in a crowded card room, which would take care of all lint; but the body of air that would have to be handled by such a system would require a large expenditure of power, to say nothing of the rather heavy cost of hoods, ducts, fans and separating chambers. It would be interesting to know whether, and how far, it is economically and mechanically practicable to equip the individual machine with its own suction-cleaning device, operating as a part of the machine itself, somewhat after the manner in which the automobile motor runs its own fan for the cooling of its cylinders.

LIGHTING

31 The artificial lighting of textile mills is a field which offers great room for improvement. Electric lighting is usually adopted. There are many mills where gas lighting is still used and a few where the old oil lamp may be found. A clear white light is the best for all purposes, but with many forms of the incandescent bulb the light actually obtained is usually quite yellow. On all white work the Cooper-Hewitt light has been used with very good results, since it is steady and gives a powerful illumination. It has been found very serviceable in machine shops, and it has the virtue of allowing one to concentrate his gaze without undue fatigue of the eyes. In cotton mills, however, the operatives generally prefer local to general lighting, but this involves many difficulties in securing the right arrangement of the different lights. The placing of a hooded bulb close to that part of the machine which the operative must keep under his eye has been one solution of the difficulty of cross shadows which arises when a series of lights are placed near a row of machines. The hooded local light, while it seems at first sight desirable, probably tends to provoke accidents, owing to the deep shadow surrounding most of the machines; and the alternation of the operative's gaze between the brilliantly lighted patch of his machine and the comparative darkness which surrounds that lighted patch has been found

to be very hard on the eyes. This would seem an inevitable result, because a quick glance from a brightly lighted spot to a dark spot and then back again to the light spot involves two very rapid accommodations of the eye mechanism, and a consequent excessive strain. The artificial lighting of a textile mill is in fact almost a special field of engineering.

WOOL AND WORSTED MANUFACTURE

32 While much that has been said of the cotton mills applies as well to woolen and worsted mills, there are many problems that are more or less peculiar to woolen and worsted mills. Some of the more important of these are here noted.

33 Machinery offers a great field for improvement. One of the greatest present wastes comes in the extremely short life of the leather aprons and dabbing brushes on worsted combs. The annual cost of renewals of the leather apron which carries the stock to the combs in a large mill may easily exceed $10,000. Metal rolls have been tried as a substitute for the leather apron, but without success. The problem is as old as the industry and it would seem that a solution should be forthcoming. The ineffectiveness of many devices tried in the past raises the question whether an entirely different type of machine is not needed. Dabbing brushes often wear out in about two days, and apart from the expense of the new brushes themselves, the bother of the whole operation is a great nuisance. Something new is needed here, possibly something new in principle.

34 The French system of spinning raises another serious problem in the difficulty of getting the right conditions for the best handling of the fiber. There is much electricity. Proper conditioning of the air for the fiber demands both high temperature and high humidity. The crux of the whole difficulty seems to be in the heating of the rolls of the machine, the temperature of which is the most important single item in successful spinning. They must have the same temperature as the air of the room; if they are only a little cooler spinning goes badly, while if they are comparatively cold, spinning is practically out of the question. The problem here is whether the roll can be heated to an efficiency temperature by some internal application of heat, so that its temperature shall be practically independent of the air.

35 A variety of other problems may be grouped together, not because they are individually unimportant, but because there is no need of enlarging upon them after what has already been said in more

detail of the problems of the cotton mill. Dust in the singe-room is one of these problems. This dust falls to the floor and can be cleared away with an air blast or air suction rather more easily than cotton lint; but some kind of continuous removing process is necessary because of the extremely irritating effect of this dust when breathed into the lungs. Sulphur dioxide gases in the bleachery offer another problem in removal; a somewhat similar problem arises also in the carbonizing room where the treatment of wool with acids gives rise to very destructive action upon shafting and wood work. Ventilation of hot-rooms and dye-houses is another removal problem. These air-borne by-products of the woolen industry, comprising as they do, steam, various destructive gases, dust, an excess of heat and humidity, might seem again to furnish material for a special branch of engineering. Proper utilization of the waste products from wool washing presents another opportunity for the engineer. In regard to machinery, there is at present much trouble from soiling of fabrics by the oil used on the machines through which they pass, especially in the finishing machines. Such stains are of course disastrous on white or light-colored goods and are more serious than similar stains on cotton fabrics because of the much greater value of the material itself. There seems to be in the woolen manufacture the same question that arises in respect to cotton, whether the whole machinery of manufacture cannot in some way be made to give more output and better product by some departure from traditional methods. Existing processes leave something to be desired, not only on the side of their producing efficiency and maintenance cost, but perhaps quite as much in the deterioration of the wool fiber that results from existing processes.

MANAGEMENT AND LABOR EFFICIENCY

36 There remain to be mentioned some apparently minor matters relating to the efficiency of operatives and to details of administration within the mill that are at least worth attention and that may hold the possibility of considerable economies. The influx of labor from the near East and from Southern Europe has brought mills in some sections to dependence on a class of labor whose neatness, efficiency and general adaptability to the conditions of cotton manufacture are considerably less than ideal. Certain classes of operatives for instance, who in their foreign homes never recognized the need for clean hands, bring their old point of view into the cotton mill, where in making piecings and tying warp threads with smudgy fingers they

multiply troubles for the overseer. This problem of getting the operative's hands clean enough for the handling of white goods is a tolerably serious one in some mills. Certain plants provide each operative in the spinning and weaving departments with a pail of water in which he is expected to wash his hands as often as is necessary to keep them clean. Operatives of this class will not go to set basins at considerable distance from their working stations. It might be well to have a small hand basin between every pair of windows.

37 Prevention of accidents to operatives seems to require that some special instruction should be given to this lower grade of labor before the new operative is allowed to come into the mill. It is not unlikely that if green operatives had a week of preliminary instruction in which they made the acquaintance of various types of machinery and were shown what part of the machines might lead to accidents, the mill managements might find the expense of such instructions a profitable outlay.

38 In view of acknowledged defects of certain kinds of mill labor there is the question whether employers would not find it profitable to institute and maintain some continuing scheme for the instruction of new operatives, the picking out and retaining of those operatives who showed themselves capable of high efficiency, and the rejecting of those operatives with whom the instruction did not take. Up to the present time textile improvements designed to reduce the labor cost have been in the direction of requiring less labor and some of those who are alive to the defects of some of the present labor supply, see a remedy only in making textile machinery still more automatic.

39 Finally, in many of the older textile plants of the country the typical American policy of scrapping inefficient equipment needs to be extended to the buildings themselves. The comparatively recent erection and profitable operation of various new mills located in parts of New England removed from the traditional manufacturing centers emphasizes the necessity, under the conditions of today, of having the best possible plant arrangement in the matter of buildings. Some mills have gone under and others are weak, because they have been unwilling or unable to discard wholly a plant layout which no amount of internal improvement could raise to the pitch of cost efficiency required for successful competition with new mills properly laid out at the beginning. As reorganizations take over unsuccessful properties, it is fairly certain that a considerable number of old mill buildings will be razed, so that new ones may be put on the same sites; or that the old buildings will be altogether abandoned as manufacturing plants.

DISCUSSION

Charles T. Main said that the purpose of the paper was to outline the possibilities of the work of the Sub-Committee on Textiles, and the important aid which could be given by members of the Society. There are many elements entering into the work and some knowledge of nearly all the branches of engineering is required for the execution of this work. The two principal considerations in the manufacture of textile goods are machinery which will turn out goods of a proper quality in maximum product and labor sufficiently skilled to run the machinery properly. All other problems of building construction, production of power, etc., however important, are subsidiary to these. There is always, therefore, open to the mechanical engineer the possibility of improving the design of the textile machinery.

Next in interest are problems in connection with production, purchase and distribution of power. Very little knowledge is available as to the cost of power in mills under actual running conditions, and some good work might be done in the collection of such data. These would be most valuable to a manufacturer who is in a position either to make his own power or to purchase power or electric current, but who is not sure what it would cost him by either method. There is great diversity of opinion, especially between mechanical and electrical engineers, as to the losses by these two methods. The losses shown by John T. Henthorn in his paper read before the Society[1] a good many years ago are very much less than is usually claimed by the electrical engineers. Some further data of the relative losses in modern mills would be of value. One of the greatest claims for electrical transmission is the increase in product due to a more uniform speed near the allowable maximum. Definite information along this line would be of great value as the question of whether the increased cost of electrical transmission is warranted often turns on this point.

The author makes a plea for reinforced-concrete buildings. This type or some other form of fireproof construction is bound to increase

[1] On the Power required to overcome the Frictional Resistance of Engine and Shafting in Mills, and its Cost, Trans.Am.Soc.M.E., vol. 6, p. 461.

in proportion to the increased cost of lumber and its scarcity. While he was aware, Mr. Main said, of the advantages of this style of construction, he also knew that its cost at present was somewhat more than for the slow burning type, and that some of the advantages had been overdrawn and some of the disadvantages overlooked.

With reference to glass area, the author stated that 80 per cent could be obtained with concrete construction and 50 per cent maximum in brick. In two mills with brick walls which he had recently built, the glass area was 56 per cent, and in one side of one mill, 63 per cent of the total area; and these percentages could have been increased if the sills had been put lower, as they must be to get anywhere near 80 per cent in concrete buildings.

In nearly every mill which he had known about where there were concrete wearing surfaces on the floors, there had been more or less trouble, and in many mills wood top floors were put in. There had also been trouble in expecting cement floors to be absolutely watertight, when wet work was placed on a second floor.

The author places a good deal of stress on the rigidity and consequent wear and tear on machinery. This is at variance with former opinion, that some elasticity is good for this kind of machinery. He was not sure that the author was right and would like to see some proofs. He then outlined the following field for work: To compile some data on the relative cost of various types of buildings built exactly for the same purpose; to give accurate figures relative to the percentage of glass area in the different types of construction; to devise a way to make the wearing surfaces of concrete floors absolutely sure; to make further studies of the proper kind of glass and of the best types of window to use; to study the effect of the semi-elastic construction of the slow-burning type, and the rigid type of reinforced concrete.

One subject not mentioned by the author which is worthy of the consideration of the engineer, is the heating and ventilation of dye houses and similar rooms, where a large amount of vapor is given off from the kettles and other machines. There are very few such rooms in which the conditions are satisfactory. They are apt to be filled with fog and dripping from the ceiling, and sometimes very hot. All of these conditions add to the discomfort of the operations, to the cost of production and cause more or less injury to the goods.

George R. Stetson said that in his community several new mills have adopted the electric drive with satisfactory results, but there

is also considerable question as to whether or not a great central station for the generation of power, which would send to these different mills the current required, would not effect great economy. Another phase to be considered is whether or not electric drives will increase production and improve the character of the work. A breakdown in the electrical apparatus that interrupts even for a short time an important function of a mill will disarrange the entire organization. The speaker knew of such an occurrence in a certain mill. Positive knowledge on these points would be valuable both to those who furnish the current and to those who receive it.

JOHN ECCLES[1] agreed with what had already been said as to electric drives being one of the important factors in large plants; where three or four mills are closely connected (within one-third or one-fourth of a mile), it is probable that it would be of great advantage to have a central power plant.

Another question for the consideration of a mechanical engineer is an individual drive by motor, for each machine, in the different departments; at present the question is whether to have a motor to drive one self-acting mule and also one for each warp spinning-frame, or to have one motor to drive groups of several machines. There are no accurate figures available to determine which of the two systems is the more efficient in production, and more economical as to cost of installation.

Mr. Eccles stated that it is usually necessary to board over concrete floors. Another matter is to be considered in connection with mills, say 120 or 150 ft. wide; in order to have sufficient light in the middle requires large window-area of glass, say about 9 by 12 ft. Does not this require more heat to keep the rooms warm in winter at points near windows? Is this a question worth your consideration?

G. H. PERKINS said that the textile schools today are doing a great work in this industry and that there the future leaders in this work must be educated. As a representative of the Lowell Textile Schools, he could say that the schools would be very glad to coöperate with the Textile Committee of the Society in any possible way that might further the work and to aid in securing valuable information along the lines mapped out.

[1]Taftville, Conn.

CHARLES FELTON SCOTT said that the reference to electric drive was one in which he was particularly interested as it had received more or less of his attention for some time past, and that he would like to add one or two things in connection with the points already made. At a convention of the American Institute of Electrical Engineers a year ago he met the officials of the Southern Power Company and learned the relation of the company to the cotton mills in that district. Its lines are extended over great areas reaching hundreds of miles in every direction, forming a general power system. The old mills were equipped with their own water powers or steam plants. Some new mills installed the electric drive and it soon began to extend to other mills. A curve was shown indicating the increased cotton mill load for several years. It took, he thought, two or three years to get it up to 10,000 h.p., and in the next two or three years it had run from 10,000 to 100,000 h.p., the curve running up very sharply, and at the end of that time something over 100 mills of the 150 in that district were supplied with this system.

The advantages found in the electric drive are not merely confined to the cost of power, but also lie in the excellence of the power. It was desired in these mills to maintain the maximum practical speed at all times. With the steam plant this maximum speed was secured in the morning, but in running over the day the belts became softened and the slippage was greater, and the machines furthest from the engine were running several per cent low at the end of the day, giving an average speed throughout the day of some 2 or 3 per cent less the maximum in the morning. With the electric drive, the maximum speed was obtained continuously and an increased output of several per cent was gained, in itself about equal in value to the cost of the electric power.

The quality of the power, its uniformity and steadiness, flexibility and reliability, the improved quality of the output, and the advantages of adaptability in making arrangements in the mill, are sometimes overlooked when the mere power bill is counted and compared with the cost of coal for running the steam engines.

The business of the mill owner and operator is to run a mill and not to make power, which has come to be an art in itself. The machinery, the methods of operation, the economies which can be secured in power manufacture by power experts, are in a way on a par with the expert mill operation. Why not, then, get rid of the power appliances, the running of engines and all that sort of thing as much as possible, and let those matters go to the experts in the power company capable of handling work of that kind?

In a visit to the city of Hartford, he had made, he said, a little study of the Hartford plant. This is a manufacturing city and a number of the prominent industries located there, making products involving the expenditure of millions of dollars, use power from the central station company. There are practically no individual plants not using this power, which seems to be an answer as to whether central station power is advantageous. The manufacturers for ten miles in various directions from the central plant are discarding their steam plants, even in factories which use a good deal of waste steam for heating purposes, and are buying electric power, not because it is cheaper but because it is better.

In a test made in a New England machine shop not long ago of group drive versus individual drive, on one floor of the factory involving some 130 or 140 different machine tools, driven from one source of power, the engine, the question was whether it was better to put in a 150-h.p. motor to drive that floor or about 130 motors of 2 h.p. each. The cost of the one motor was less than $500, while that of the 130 motors was over $8000, more than sixteen times as much. It seemed at first absurd to put in the little motors, but when the cost of the shafting and belting was included with the cost of the large motor, the total first cost of the two installations was practically the same, within a per cent or so. Furthermore, the cost of the power lost in the shafting and belting amounted to about $1700 per annum, which is a pretty good interest on the first cost of the investment.

The total cost of power in the mill, however, is usually only a few per cent of the total cost of the product, and the discussion as to whether one kind of power or another is going to involve the greater cost is simply a decision between, say, 3, or 3.5, or 4 per cent of the total actual cost. If in the beginning electric power were selected, there is certainly a better and larger product, with more reliability in operation, and with less attention needed from the managers of the plant. The gain may possibly amount to 5 or 10 per cent, so that the mere cost of the power should not overweigh these much more important things. The suggestion that this whole matter could well be reviewed by a committee is a most excellent one.

Arthur J. Herschman said there are two important reasons not mentioned so far, which should be in favor of the electric drive as compared with the mechanical drive. With the electric drive there is no obstruction of light, a feature that is very noticeable in textile

factories having belt drives. Another important consideration is the fact that with electric drives the danger from belts hanging down from the ceiling is avoided. The question of electric versus mechanical drive depends primarily upon the kind of plant under consideration, whether it is purely a spinning plant, or a combination of a spinning and weaving mill, and upon the general layout of the plant. Where it is possible to have the boiler plant in the center, with the engine room ahead of the boiler plant and two large wings extending right and left, as is the usual practice in continental Europe and also in the cotton district of England, it is a very easy arrangement to drive with ropes and, he believed, for economy, it will take a long time to improve on rope drives.

In England, a common way of driving cotton mills is by what is known as the king shaft, a powerful vertical shaft driving bevel gears right and left into the room. With small mills, the frictional waste of mechanical driving very frequently runs into 30 or 40 per cent of the total power consumed. He had tested mills some years ago of which the frictional loss was 30 per cent and more.

One disadvantage of the wooden floor is its great fire hazard. In cotton and textile mills, oil is generally used to a very great extent, and when the wooden floor is oil soaked, the oil saturates the wood, becomes imbedded in it and cannot be removed. On the other hand, in one textile mill it happened that the engine was seriously affected by the softening of its concrete foundation.

James A. Pratt believed a great deal might be accomplished by investigating the possibilities of belt drives. The Society has at its disposal exceedingly valuable material presented by Mr. Barth and Mr. Taylor, and if the designs given by these gentlemen were followed, the efficiency of belt drives might be very greatly increased. No other line of machine designing presents so many contradictions and varieties of method as in the design of cotton machinery. A great deal is left to judgment, to chance and to the advice of others not mechanical engineers, instead of carrying out a regular design development. Advocacy of the latter course on the part of members of the Society would be decidedly helpful.

A possible objection to the group drive is that it is difficult to keep separate belts tight running to a machine. This objection is not very serious because it is possible to use a hook adjusting shipper, by which a belt is adjusted by an idler running over its face and set to different positions by a hook, which

is nothing but a rough casting and very cheap. By this means, the belt may be kept at any required tension, and efficient results secured. A previous speaker mentioned the obstruction of light by the belts. This objection may be largely overcome by using an under machine drive, coming from the floor below up to the machine. Unless it is desired to use the ceiling of the factory for reflecting purposes, this arrangement interferes with the light no more than motors and individual drives.

In connection with the comparative advantages of the individual motor over the belt driven steam plant, it is well to remember that with electric drives it is necessary to have a man to look after the electrical equipment. In the case of belt drives, this feature of an attendant is too often neglected, to the detriment of the system. There is much to be said in favor of the belt drives when properly designed, installed and operated.

J. Irvine Lyle said that he had been waiting for some time for some one to discuss this question of control of temperature and humidity. Those connected with textile mills realize that there is little trouble in controlling temperatures in winter when they are apt to become too high rather than too low, especially in spinning mills, due to the immense amount of heat developed by the friction of machinery, so that Mr. Eccles' remark as to the cost of heating these buildings with large window surfaces would apply only during the night, Sundays and holidays, not to the working hours. It is perfectly possible to maintain constant temperatures in a mill throughout the entire winter within 2 or 3 deg., and to control the humidity within 2 or 3 per cent, and then to control both automatically.

In summer, conditions are not quite so easy, due to the fact that refrigeration is out of the question on account of the immense cost; but it is possible to keep them at much lower temperatures than could be done without a cooling system. By the evaporation of water it is possible to get a reduction of temperature varying from, say 12 deg. to about 25 deg. below the outside temperature, and then by introducing into the mill a volume of this cooled air sufficient to absorb the heat, the temperature of the mills can be greatly lowered.

He knew of one mill, he said, where there are two separate buildings, one with 16-ft. and the other with 18-ft. studding, the latter having proportionately more glass surfaces. The same class of work is being done in the spinning rooms of both, and they are equipped with the same type of drive, so that as nearly as possible

conditions are equal. In hot summer weather the temperature in the one into which the conditioned air is introduced can be kept down to an average of about 12 to 15 deg. below the temperature of the other mill, and at the same time the humidity can be held up and maintained constant at the same point, so that the work runs much better.

Mr. Reynolds speaks of the slowness of production in the first hours of the morning, due to the fact that the conditions have been allowed to fall off during the night, the buildings being too cold or the humidity running too high or too low. All mill men know that their worst troubles come in the first hour of the morning. In the first hour Monday morning less production is obtained than in any other hour in the week, largely due to the fact that conditions have changed during the night. To work the fiber properly, warm temperatures and certain humidities are needed for each different process. By maintaining these constant and uniform at the same point, a greater production can be obtained at a cheaper price and with a better standard of quality than is possible where the conditions vary. He knew of one mill which does weaving, where second-grade raw material is now bought since apparatus was put into operation whereby constant conditions could be maintained. This grade can be bought for 10 per cent less than the better grade, and the product is fully as good, if not better, than that turned out before; that is, it is fully as good as could have been secured with the better grade of material without the proper air conditions.

This is a field that the engineer can study with profit both to himself and to the mill.

Calvin W. Rice stated that he had had a part in the final engineering application of electric drives in most of the industries and particularly in the textile industry, and in connection with some of the largest and earliest plants in America had attended to the various features, from the engineering to the installation of the apparatus and the operation of the plant. He was able to sell to certain plants by virtue of the merits of the electric drive, and would be pleased to supply the specific data which have been called for, showing the exact improvement in output over a long period of time, exact improvement in quality, due to steadiness, and those other features which are essential to the comparison of the electric drive with others used in mill operation. Speaking as secretary of the Society, Mr. Rice further thanked the committee for this paper which he regarded as an ideal production for a special committee.

GEORGE M. BRILL stated that there were one or two points which he had discovered in connection with other lines of manufacturing which he thought might apply with some force to the use of motors in cotton mills. In a number of cases where electric power was applied to mills previously equipped with mechanical drives, the product was very materially increased. In the Studebaker Manufacturing Company of South Bend, Ind., they were running along in a very comfortable way with a large percentage of the total fuel as refuse, quite satisfied with the product, but for certain reasons thought it best to change over to electric drives. Tests indicated very clearly that it was entirely possible to materially increase the products with less power. The workmen thought they were going through the movements of production as rapidly as possible, but it was found as a result of the installation of the electric drives that they were feeding through the material at more rapid speed and the products were materially increased without increase in the cost of labor.

Another important consideration is that of safety. With long lines of shafting and large flywheels the momentum of the parts is so great that it is impossible to shut the plant down instantly or in a few seconds; while with electric motors it is possible to stop the machinery in very much less time than in the case of a mechanical drive. An accident which might otherwise be serious, is often reduced to a minor event by this fact. Another point which applies with some force to cotton mills is the stirring up of the dust due to the large number of pulleys and heavy belts, which is, of course, greatly reduced by the use of motors.

DWIGHT SEABURY said that he was designing some mills in Vermont in which electric drives would be used for all machinery. In his opinion the electric drive was superior to the belted drive, but had the disadvantage of increased first cost. If the cost of motors and the cost of power were as low as they should be, all mills would use the electric drives.

CALVIN W. RICE remarked that the cost of power was a question of engineering. Almost the entire industry of the city of Chicago, including transportation, lighting and operation of hotels, etc., receives its supply of electrical power from one central station and a substation. Power is being produced in Chicago at less cost per kilowatt than it can be purchased in Buffalo from Niagara Falls.

G. H. Perkins expressed his belief that the entire question of electrical versus mechanical drives resolves itself in the one of cost, selling prices, and gross profits, and that in his opinion the ideal opportunity for central station power drives is to be found where a number of plants of small units lie close together. In Chicago they are producing electricity at less cost than anywhere else in the country and it must be that they are selling it at a price that the manufacturer can afford to pay. This is a good example for other central stations to follow so that they may get their price down to a figure within reach of the manufacturers of textile materials. Another thing to be considered is the geographical location of the various plants. In New England where an auxiliary plant must be installed to take care of the heating and process work, it has an effect on the cost of power for manufacturing. The South gives a different problem for consideration.

Mr. Perkins expressed his belief that the interest aroused by the development of the electric drive because of its comparative newness, and the aggressiveness with which it has been put before the manufacturers, had made too great an impression upon mill owners. In his opinion, if equal interest were taken in the development of other prime moving apparatus, it might be possible to develop it to such an extent that it would compete successfully with the individual electrical drive. In applying the electric drive to textile plants, each mill must be considered individually, and that is where the engineer should take up the problem and endeavor to meet the requirements of the particular plant in the best possible manner. Upon how well he does this depends his success as an engineer.

George M. Brill answering a question of Mr. Edward Robinson as to the comparative cost of electrical power in Chicago and Buffalo, said that although he could not give the exact figures at which the Edison Company was selling power in Chicago, the rate was sufficiently low to induce the surface and elevated railroads to take power from it and to shut down their own plants. It was also low enough for the use of a large number of manufacturers and a still larger number of isolated plants of all kinds, department stores, hotels, etc. In this connection he observed that the Milwaukee Electric Railway Light & Power Company buys power developed from water power by the Southern Wisconsin Power Company, located at the falls of the Wisconsin River about 110 miles from Milwaukee. The best rate that the water power company could give was not sufficiently at-

tractive to induce the railroad company to run their entire system of electric cars on this power. They could make power themselves as cheaply, and although they are now buying a portion of their power, they are making the bulk of it themselves from coal.

E. D. Dreyfus (written). Textile engineers deserve a great deal of credit for the manner in which they have induced means for the economical supply of their combined power and heating demands, which must be properly taken care of as previous discussors noted. To them is primarily due the development of a special design of turbine, known chiefly as the "bleeder" type, which automatically delivers low-pressure steam and electric power under the most efficient conditions of operation. This particular design of turbine differs from the regular construction mainly in that an automatic valve is arranged between the intermediate and low-pressure stages, which regulates the flow of steam to the final stage of the turbine; or, in other words, admits to the low-pressure stage all surplus steam issuing from the high-pressure sections which is not required for heating purposes. Hence it will be appreciated that there will be no waste of exhaust steam to the atmosphere at any period, and thus all steam generated will be utilized to the best advantage. That they were the first connected with this advance in turbine design illustrates the progressiveness of the textile mill directors; and it is worthy of note that all other industries in which a combined lighting and heating load occurs, have since similarly recognized the merits of having an efficient machine which will automatically serve the variable steam and power requirements.

Albert L. Pearson (written). In this paper very little reference is made to the electrical engineer and what has been accomplished in the development of motors for all systems of electric driving, and the lighting situation. I believe that a great deal of responsibility for the successful operation of a mill rests with the electrical engineer.

Motors for use in systems of group driving can be obtained which are well suited to meet conditions and which have excellent operating characteristics. Motors can also be obtained for systems of individual driving which have been developed to meet the special needs of machines for practically all of the processes. Those on the market today are the result of a careful study of conditions and tests of power required, and have also very good operating characteristics, with the possible exception of some of the smaller sizes in which the efficiency is low.

There are a number of advantages attending the use of electric drive, some of which have been mentioned in the paper. In the case of the individual system there are some others which might well be noted:

a More efficient results in illumination, both natural and artificial, due to absence of overhead shafting and belting.

b Cleaner and more sanitary rooms, for the same reason.

c Elimination of danger to operatives from shafting and belting.

d Elimination of cost of power for a given section when that particular section is not in operation.

There are several points to be borne in mind in selecting an equipment of motors to which I will also call attention:

a The efficiency and power factor should be high, this latter having considerable bearing upon the regulation of the systems generally. In case of a large installation, this characteristic may result in some saving in copper. The curves of efficiency and power factor should be as nearly flat as possible, in order to insure the most economical operation at loads ranging, say from 50 to 125 per cent of the rating of the motor.

b The percentage of slip or difference between synchronous and full-load speeds should be small to insure as nearly as possible constant speed at the machines under all conditions of load. This point is of special value in cases where one motor is used to drive four spinning frames and with group drives for looms and other machinery where close speed regulation is not only desirable but often necessary. In the individual system this is not of so much importance, as the driving mechanism will be designed to meet the full-load speed of the motor.

c The air gap should be reasonably large which will allow a longer life to the bearings and minimize the amount of attention at this point.

d Ventilation should be good and the ducts for this purpose readily accessible for cleaning. For small motors for individual driving it is better to rely upon radiation for carrying off the heat, and thus eliminate ventilating ducts.

e For textile plants an overload capacity in a motor greater than the standard 2-hr. rating is not necessary. The cost of a motor with a rating of 125 per cent load con-

tinuously is an unnecessary investment and the result of such a condition will be a continuous operating loss.

f Auto-starters and oil switches should be as simple as possible, consistent with good operating characteristics. Contacts should be liberal and so designed that repairs may be made readily with a minimum amount of time and expense.

g A plant should not be "over-motored," as such a condition is sure to result in poor operating characteristics generally. It is better to operate an induction motor at a slight overload than at an underload.

h The power supply for a mill should preferably never be less than 550 volts, 3-phase, 60 cycles, except possibly where motors smaller than 1 h.p. are used in which case better results may be secured from 220 volts. Very satisfactory results are being obtained from 2200-volt motors.

The frequency of 60 cycles is mentioned, as the range in speeds is greater than that of 25 cycles and other frequencies, and where lighting is supplied directly from the power lines more satisfactory results are obtained.

In an electrically driven plant the wiring takes the place of shafting in a mechanically driven one. This should be installed in a thoroughly first-class manner, protected from injury and designed to maintain constant rated voltage at the motors. Fuses should be eliminated, as far as practicable, and where required, should be placed in readily accessible locations.

Finally, in selecting an equipment the reliability of the apparatus and economy of operation should be given first consideration.

I agree with Mr. Reynolds that the artificial lighting of mills is a field which offers great room for improvement, and that it is, in itself, a special branch of engineering.

Electric light is becoming the standard, generally, as it is the most convenient, economical and safest illuminant available.

There are two principal forms of electric lamps, arc and incandescent, of which there are several types. Those in most common use in mills are as follows: arc—enclosed carbon, mercury; incandescent—tungsten, tantalum, carbon.

There are a great many systems of arc lighting in use and some of them are quite satisfactory. The mercury arc is used to a certain extent and with very satisfactory results in mills making white goods exclusively.

The tungsten lamp is the most efficient of the incandescent class

and in its present high state of development is practically free from the early objections to its use. In addition to its efficiency the quality of the light is vastly superior to that of other illuminants, approaching most nearly the ideal. This lamp has revolutionized the lighting situation and is being adopted as the standard of best practice for the incandescent system.

During the past few years more attention and study have been given to the proper arrangement of lighting systems. Reflectors have been developed to meet special needs and as a result the more recent installations are superior in every way to the older ones.

There is, however, a great diversity of opinion among the mill owners as to the best system to adopt and, as stated before, there is room for great improvement.

The operative should be taken into account in developing any system of illumination. As the light is provided for his use, his needs should be given every consideration.

Lamps hanging low, which can be adjusted by the operative, should be avoided. Not only is he liable to experiment with the lamp in this position and waste time, but he may also interfere with the light and work of other operatives. The lamps should be so arranged as to give uniform illumination, avoid shadows as far as possible, and particular attention should be given to the requirements of each machine.

White walls and ceilings are advantageous and add to the effectiveness of the distribution of light. In cases of systems of individual drive it is possible to keep the walls and ceilings cleaner than with the group system or mechanical drives.

On account of glare, low exposed units should be avoided. Where it is necessary to have the light units low, reflectors should be used which will entirely conceal the filament of the lamp. In such cases, it is usually desirable to have in addition a number of units hung close to the ceiling to provide illumination for shafting, where there is such, and to overcome the effect of light and darkness.

The distributing system should be carefully worked out in order that the voltage regulation may be good. Circuits and switching should be arranged with respect to the different processes in such a way as to eliminate the use of power for lighting in sections which are not in operation. This should also be laid out so that the central portions of the mill may be lighted independently of the portions next the walls. This is particularly important in wide mills, but does not apply, of course, to upper stories or to one-story buildings with saw

tooth skylights in which light is required all over the room at practically the same time.

CHARLES T. PLUNKETT, chairman of the meeting, said that in closing he would like to call attention to several paragraphs which might be of some assistance in the consideration of the paper.

As to the importance of the textile industry, shown in Tables 1, 2 and 3, taken from the census of 1900, and for more recent figures, approximately 20 to 25 per cent may be added to the increase of the spindleage in Great Britain, and in the United States about 40 per cent. The spindleage of Great Britain is not languishing, where within 40 miles of Manchester, in and about Lancashire, there are twice as many spindles as in the United States. In Russia there are now nearly one-quarter as many spindles as in the United States, and in visits to some of these mills and to those in the several countries of Europe, almost invariably the wages paid in Russia were found to be practically one-quarter of those paid in America, from the power plant through to the finishing room. In Italy it is about one-third, in Germany 50 per cent, and in Great Britain about 65 per cent. This does not indicate exactly the difference in cost, because, with the exception of Great Britain, there is less machinery attended by an operative. In Russia, two looms seem to be the limit of capacity or willingness, and the same is true in Italy. However, in the latter country a modern mill was found entirely of concrete construction, with a roof of saw tooth type, in which there were 500 German looms and 500 of the most approved automatic looms, 8 of the latter being run by each weaver.

In Germany, and in all of these countries, in fact, the carding and spinning machinery is largely purchased in England. Lancashire manufactures nearly all of the machinery of this class used in Europe and Asia.

Mr. Reynolds has spoken of the disappearance of the native American labor. That was very true in this industry and he presumed in all others. Whereas a half century ago the labor was American, the German and Irish laborers came in in the fifties and sixties and were the principal foreign-born operatives at that time. The French Canadians followed, and since that time, Poles, Greeks and Portuguese have largely replaced those of other nationalities. This is a disadvantage in that those accustomed to work in textile mills have the slow motion of the continental textile worker and it is very difficult to accelerate that motion. In England the work is nearly as rapid as here, both in the speed of machinery and in the

movements of the operatives employed. The more recent employee is also less interested in the success of his employer than his predecessors were.

With regard to the arrangement of machinery in buildings, he said he had recently visited a mill where the machinery had been laid out in the card room, completely new within three years. By a rearrangement of that machinery on the new floor space, there were 20 per cent less operatives required, which illustrates what may be done by the engineer.

There are opportunities, as the author says, to combine some of the processes. An interesting machine seen in Germany eliminated the spooling and made the cost much less than with the spooling process. The lint and bits of leaves were thrown from the yarn by the ballooning of the thread between the bobbins and the warper creel rings. The warper was speeded two and one-half times as fast as any warper built for spools could run.

The Germans have two or three distinct advantages over us, their inherited notions of economy, and the benefit of numerous technical schools throughout the country. Hundreds and thousands of young men are being turned out every year who know how to study, to investigate, to make researches, and with their abounding tenacity, to make a success of their undertakings. For these reasons the Germans are leading in many of the engineering callings as well as in the industries, and in none of them more than in the providing of satisfactory machinery for the manipulation of cotton waste. This loss is the most serious that the mills have to contend with and no other country has made such progress as Germany in this direction. Millions of pounds of cotton waste are being exported by this country to Germany each year, to be made into fabrics, some of which come back again. England is exporting twice or three times as much, and is just waking up to the fact that the Germans are meeting them in all the markets of the world with a fabric made from their own waste. The character of the waste finally shipped from the German mills is very nearly a fertilizer; there is scarcely a fiber in it which would be of any use. The development of machinery for the utilization of what is now cotton mill waste is a broad and important field.

The managers of the textile industries do not profess to have such technical knowledge as the engineers, but have nevertheless a practical knowledge of their power plants, the operation of which is their third largest expense.

The author did not desire to present a closure.—EDITOR.

No. 1340

RATIONAL PSYCHROMETRIC FORMULAE

THEIR RELATION TO THE PROBLEMS OF METEOROLOGY AND OF AIR CONDITIONING

By Willis H. Carrier, Buffalo, N. Y.

Associate Member of the Society

A specialized engineering field has recently developed, technically known as air conditioning, or the artificial regulation of atmospheric moisture. The application of this new art to many varied industries has been demonstrated to be of greatest economic importance. When applied to the blast furnace, it has increased the net profit in the production of pig iron from $0.50 to $0.70 per ton, and in the textile mill it has increased the output from 5 to 15 per cent, at the same time greatly improving the quality and the hygienic conditions surrounding the operative. In many other industries, such as lithographing, the manufacture of candy, bread, high explosive and photographic films, and the drying and preparing of delicate hygroscopic materials, such as macaroni and tobacco, the question of humidity is equally important. While air conditioning has never been properly applied to coal mines, the author is convinced that if this were made compulsory, the greater number of mine explosions would be prevented.

2 Although of so much practical as well as scientific importance the laws governing many of the phenomena of atmospheric moisture are but partially understood, while the present engineering data pertaining thereto are both inaccurate and incomplete. Accepted data used in psychrometric calculations are based largely on empirical formulae, which are incorrect as well as limited in their range. Recent investigators have determined the most important properties of water vapor with final accuracy. At the same time, sufficient error has been shown in previous steam data, especially at atmospheric temperatures, to warrant the revision of all calculations based thereon.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

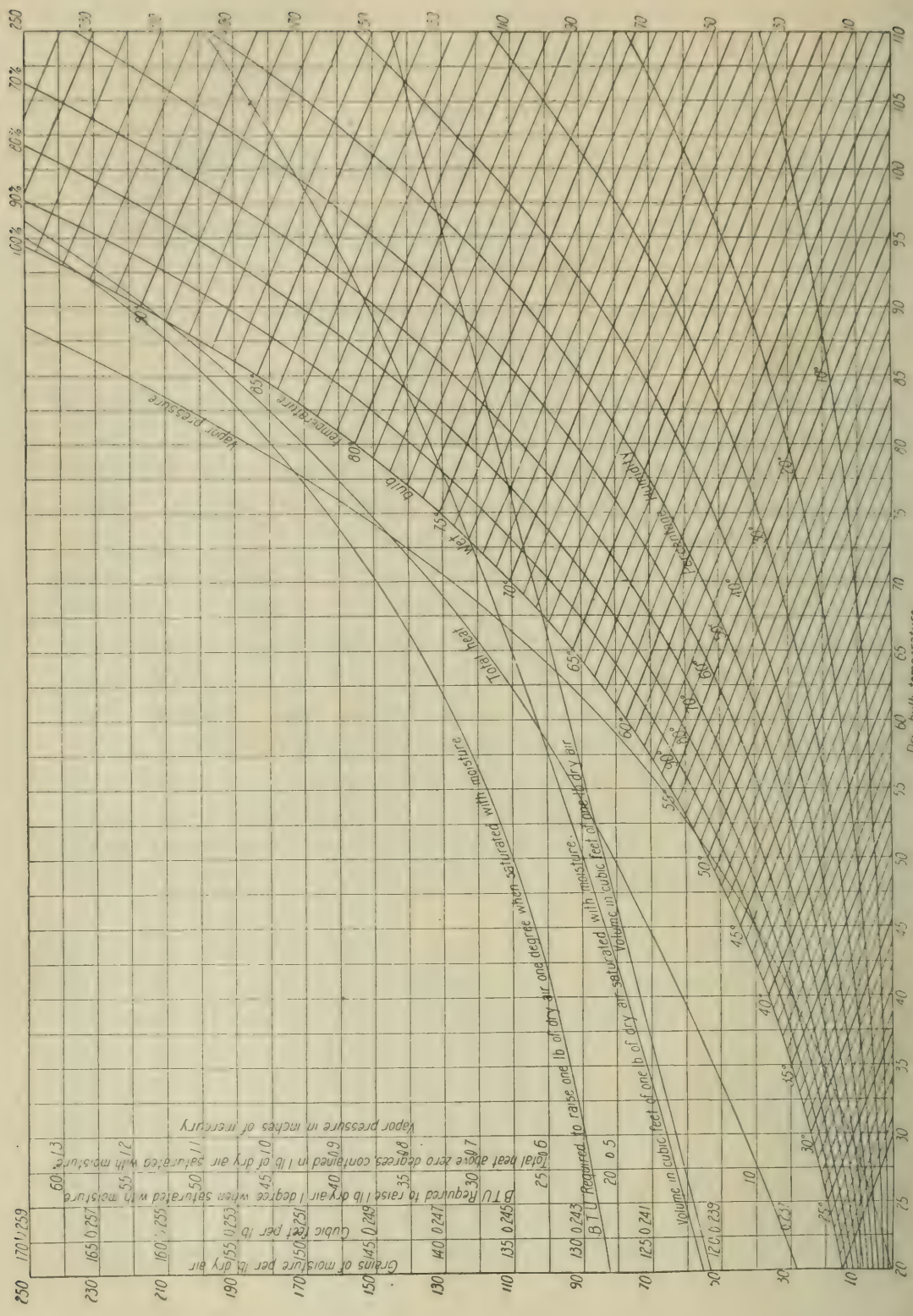

Dry bulb temperature
Wet bulb temperature
Vapor pressure
Total heat
Relative humidity
Volume in cubic feet of one lb of dry air saturated with moisture.
B T U Required to raise one lb of dry air one degree when saturated with moisture.
Total heat above zero degrees contained in 1 lb of dry air saturated with moisture
Vapor pressure in inches of mercury
Grains of moisture per lb dry air
Cubic feet per lb dry air
B T U Required to raise 1 lb dry air 1 degree when saturated with moisture

FIG. 1 PSYCHROMETRIC CHART

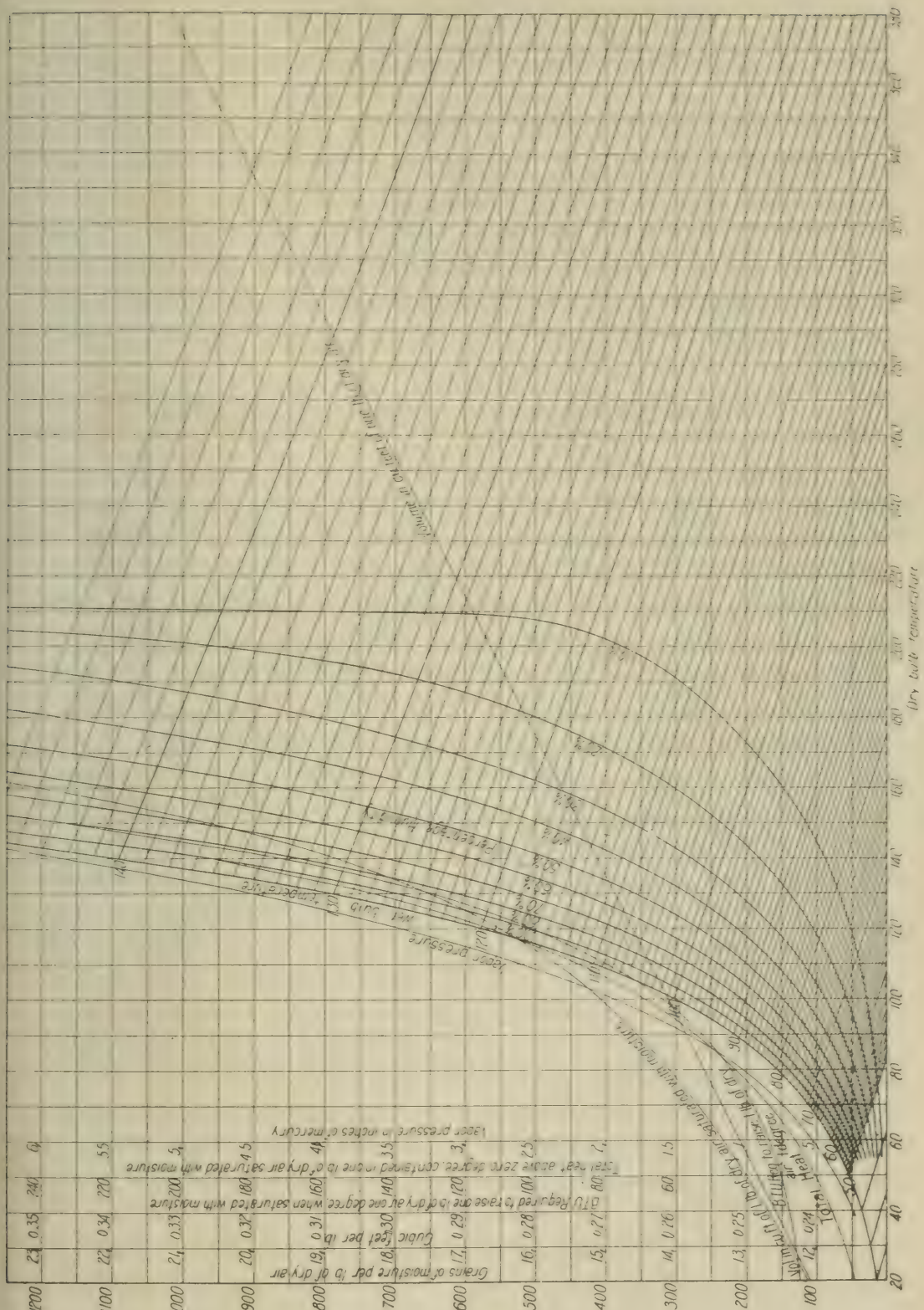

Grains of moisture per lb of dry air
Cubic feet per lb
BTU Required to raise one lb of dry air one degree when saturated with moisture
Total heat above zero degree contained in one lb of dry air saturated with moisture
Vapor pressure in inches of mercury
Weight in lbs of one cubic foot of dry air saturated with moisture
Total Heat BTU in 1 lb of dry air
Wet bulb temperature
Dry bulb temperature

FIG. 2 PSYCHROMETRIC CHART

3 It is the purpose of this paper to apply these final data to the development of rational formulae for the solution of all problems pertaining to the phenomena of atmospheric moisture as related to psychrometry and to air conditioning. Original data are given in proof of fundamental relations as well as in determination of errors in standard psychrometric instruments. The author hopes these results may prove to be of permanent value.

4 In order to establish a logical basis for the presentation of these data and the derivation of the rational formulae, the established principles and laws governing atmospheric moisture will be reviewed and the present methods of determining atmospheric humidity discussed.

VAPOR PRESSURE AND LAW OF PARTIAL PRESSURES

5 Water vapor exists in the air purely as a mixture in relation to its other elements. This vapor, according to Dalton's law, is capable of exerting a certain maximum vapor pressure dependent entirely on its temperature and regardless of the presence of other gases or vapors. For example, assume 1 cu. ft. saturated with vapor of alcohol at 100 deg. cent. having a vapor pressure of 1697.6 mm., and add isothermally to this 1 cu. ft. saturated with water vapor at 100 deg. cent. having a vapor pressure of 760 mm. This will give 1 cu. ft. of the mixture saturated with both water vapor and alcohol vapor at 100 deg. cent., having as a total pressure the sum of the two separate saturated vapor pressures, or 2457.6 mm. Similarly, an equal volume of a third saturated vapor might be added without affecting the other two. But if, on the other hand, it is attempted to include isothermally an additional amount of either of the saturated vapors, a corresponding condensation of the particular vapor added would result. In the same manner, an unlimited amount of a gas, such as air, could be added isothermally to a cubic foot of water vapor without affecting its condition of saturation, giving a combined pressure equal to the gas pressure plus the vapor pressure.

6 The established temperature-pressure relationship of saturated water vapor is shown by curve (1) on the charts, Figs. 1 and 2. This is the well-known temperature-pressure curve of steam.

PARTIAL SATURATION

7 When the temperature of a definite weight of saturated vapor is increased isobarometrically, it is said to be superheated. Its

specific volume is increased, in accordance with the law of gases, in direct proportion to the increase of absolute temperature, while its density is changed in an inverse proportion, as shown in Fig. 3; that is, $\frac{D_2}{D_1} = \frac{T_1}{T_2}$, where D_1 and D_2 are the densities corresponding to the absolute temperatures T_1 and T_2, respectively, and $(T_2 - T_1)$ is the degree of superheat. If D'_2 is the density of saturated vapor at temperature T_2, then the ratio $\frac{D_2}{D'_2}$ is said to be the per cent of saturation, or more exactly, the per cent of isothermal saturation. When these relationships are considered with respect to water vapor in

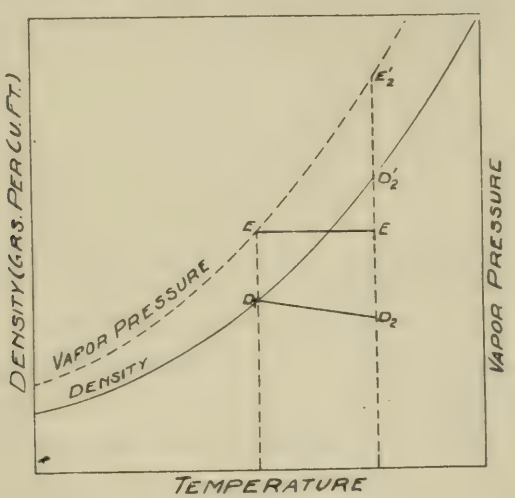


FIG. 3 TEMPERATURE-DENSITY DIAGRAM

air, this ratio is termed the per cent of relative humidity, while the densities D_1, D_2, D'_2, etc., customarily expressed in grains of moisture per cubic foot, are termed absolute humidities.

DEWPOINT

8 It should be noted that although the total weight of the water vapor remains the same, the absolute humidity D_2 is less than the absolute humidity D_1. However, if water vapor, or air containing water vapor, having a temperature T_2 and an absolute humidity of D_2, be cooled to T_1, it will become saturated, and if cooled further, moisture will be precipitated. Therefore T_1 is termed the dewpoint of air having a temperature T_2 and an absolute humidity D_2, or a corresponding relative humidity $\frac{D_2}{D'_2}$. Therefore, the dewpoint may

be defined as the minimum temperature to which air of a given moisture content may be cooled without precipitation of moisture.

9 Usually it is more convenient to determine the absolute and relative humidities from the temperature-pressure curve by comparing the vapor pressures. The per cent of humidity is $\frac{D_2}{D'_2}$, but it may also be shown to be equal to $\frac{e_1}{e'_2}$, i.e.

$$\text{Per cent humidity} = \frac{e_1}{e'_2} = \frac{D_2}{D'_2} \quad \ldots\ldots\ldots\ldots\ldots [1]$$

where e_1, is the pressure of saturated vapor corresponding to the dew point T_1, and e'_2 is the vapor pressure at saturation corresponding to temperature T_2. It also follows that

$$D_2 = D'_2 \times \frac{e_1}{e'_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

METHODS OF MEASURING ATMOSPHERIC HUMIDITY

10 Determinations of atmospheric moisture may be made by four distinct methods, chemical, hygroscopic, dewpoint, and evaporative or psychrometric.

11 *Dewpoint Method.* The dewpoint method was first brought into use by Daniels and by Regnault, and adopted by the United States Weather Bureau in the determination of the values used in their psychrometric tables. The dewpoint is measured directly by observing the temperature at which moisture begins to form upon an artificially cooled mirror surface. Determination by this method is extremely delicate and when suitable precautions are taken, is considered very accurate. However, it is questionable whether the true dewpoint is ever quite as low as indicated by this method. The temperature is usually taken by a thermometer placed in a thin silver tube filled with sulphuric ether or other volatile liquid, which produces cold by evaporation. The temperature of the exterior of this tube is undoubtedly at the true dewpoint, but it is questionable whether the thermometer at the center of the tube registers this dewpoint with absolute accuracy. The exterior surface of the tube must often be cooled 25 or even 50 deg. below atmospheric temperature in order to reach the dewpoint.

12 In any case a considerable quantity of heat must pass through the tube to the cooling medium from the external air by convection,

and to a less extent from external objects by radiation. The internal resistance to the transfer of heat of a thin plate of metal, forming the wall of the tube, is in itself negligible; however, as any one who has studied the subject of heat transmission will recognize, the surface resistances are appreciable. On the outside there is the resistance of the surface exposed to the water vapor at low tension and on the inside the more considerable resistance of the liquid surface. There is therefore every reason to believe that the interior ether is at a slightly lower temperature than the exterior dewpoint. This conclusion conforms with conditions demonstrated by other observers in tests upon the temperature of the exterior of radiating or convecting surfaces. The extreme accuracy of the results obtained by the dewpoint method at high temperatures and low humidities would, therefore, seem greatly in question.

13 *Evaporative or Psychrometric Method.* The evaporative or psychrometric method has not heretofore, to the writer's knowledge, been definitely accepted as an absolute means of moisture determination, but as will be demonstrated, is independent of and preferable to all other methods in scope and accuracy. It is of special interest in relation to the art of air conditioning, because the same fundamental phenomena are involved and subject to the same theory. It is of service not only in the art of air conditioning, but also as a departure in the science of meteorology. It provides a method, remarkable for simplicity and accuracy, for the determination of the specific heat of air, which present methods have failed to establish, within an unquestioned accuracy of 2 per cent.

14 This method of moisture determination depends upon the cooling effect produced by the evaporation of moisture in a partially saturated atmosphere. This is usually measured by covering the bulb of an ordinary mercurial thermometer with a cloth or wick saturated with water and comparing its temperature with that of a thermometer unaffected by evaporation. The covered bulb is termed the wet-bulb thermometer, and the difference between the wet- and dry-bulb readings is termed the wet-bulb depression. The temperature of the wet bulb is affected in a measure by radiation from surrounding objects. It is therefore very susceptible to air currents which serve to increase the evaporation and therefore decrease the percentage of error due to radiation. On this account, the earlier and more convenient form of hygrometer using a stationary wet bulb is very unreliable, considerable correction being necessary for radiation. The sling psychrometer advocated by the United States Weather

Bureau overcomes this error to a great extent by increasing the ventilation and consequent rate of evaporation to such a degree that the heat received by radiation becomes a small percentage of the total heat transformation.

15 The most reliable tables based on the stationary wet-bulb hygrometer are those by James Glaisher (1847.)[1] The tables of the United States Weather Bureau, based upon an empirical formula deduced by Prof. Wm. Ferrel from simultaneous determinations with the sling paychrometer and the dewpoint instrument, are more reliable and are now generally used. The limitations of this formula are admitted, since it is held to be correct only over the range of observation from which it was deduced, including simply temperatures below 120 deg. fahr.

16 Professor Ferrel's formula as given in the tables of the United States Weather Bureau is

$$e = e' - (0.000367\ P)\ (t - t')\left(1 + \frac{t' - 32}{1571}\right)$$

where

e = partial pressure of the moisture in the air, which also = vapor pressure corresponding to the dewpoint

e' = the vapor pressure corresponding to saturation at wet-bulb temperature t'

P = the barometric pressure

t = dry-bulb temperature in deg. fahr.

t' = wet-bulb temperature in deg. fahr.

17 The temperature of the dewpoint is found by selecting the temperature corresponding to the pressure e, from the temperature-pressure diagram or table. The per cent of relative humidity is $R = \frac{e}{e_t}$, where e_t is the vapor pressure corresponding to the dry-bulb temperature t, as previously demonstrated. The absolute humidity expressed in grains of moisture per cubic foot is then determined by multiplying the grains of moisture per cubic foot corresponding to saturation at dry-bulb temperature by the per cent of relative humidity thus determined.

18 The writer would substitute for such an empirical formula a rational one, having a thermodynamic basis, that is, a formula depending upon the transformation of sensible heat into latent heat in the adiabatic saturation of dry air.

[1] Phil. Trans. Royal Soc., 1851, p. 141.

19 Historically, it is of interest to note in this connection, that James Apjohn[1] propounded in 1836 this same theory of wet-bulb temperature. However, he was unable to establish the correctness of his assumptions, partly because the data then extant regarding the specific heat of air and the latent heat of water vapor were inaccurate, but more particularly because he assumed the temperature indicated by the stationary wet-bulb thermometer which he used to be the true temperature of evaporation, while as a matter of fact, it is considerably higher, owing, as we have shown, to the effect of radiation upon the stationary wet bulb.

20 The author first observed that the wet-bulb temperature given in the psychrometric tables of the United States Weather Bureau agreed substantially with the computed temperature at which air of a known temperature and moisture content would become saturated adiabatically, i.e., without the addition or subtraction of heat. These calculations were made by the writer in 1903, in determining the moisture-absorbing capacity of air in connection with the fan systems of drying. Subsequently, this relationship was still further investigated and thoroughly established in connection with the system of air conditioning introduced by the writer.

21 Tests upon progressive fan-system dry kilns in 1904 disclosed the fact that the wet-bulb temperature was substantially the same in all parts of the kiln regardless of the drop in temperature due to moisture absorption, a phenomenon which logically results from the identity of the wet-bulb temperature and the temperature of adiabatic saturation.

PSYCHROMETRIC PRINCIPLES

22 The following principles underlie the entire theory of the evaporative method of moisture determination, as well as of air conditioning:

(*A*) *When dry air is saturated adiabatically the temperature is reduced as the absolute humidity is increased, and the decrease of sensible heat is exactly equal to the simultaneous increase in latent heat due to evaporation.*

(*B*) *As the moisture content of air is increased adiabatically the temperature is reduced simultaneously until the vapor pressure corresponds to the temperature, when no further heat metamorphosis is possible. This ultimate temperature may be termed the temperature of adiabatic saturation.*

[1] Irish Academy Trans., vol. 17, pp. 275–282, 1837.

(*C*) *When an insulated body of water is permitted to evaporate freely in the air, it assumes the temperature of adiabatic saturation of that air and is unaffected by convection; i.e., the true wet-bulb temperature of air is identical with its temperature of adiabatic saturation.*

23 From these three fundamental principles there may be deduced a fourth:

(*D*) *The true wet-bulb temperature of the air depends entirely on the total of the sensible and the latent heat in the air and is independent of their relative proportions. In other words, the wet-bulb temperature of the air is constant, providing the total heat of the air is constant.*

24 A statement of the experimental demonstration of these four principles is given in Appendix No. 1.

APPLICATION OF THE EVAPORATION CALORIMETER TO THE DETERMINATION OF THE SPECIFIC HEAT OF AIR

25 In consequence of the psychrometric principles *A*, *B* and *C*, the moisture content of air from accurate psychrometric readings may be easily computed, provided the required temperature is known, as well as the density relations in a mixture of pure air and saturated water vapor, and also the exact latent heat of water vapor, and the specific heat of air and of water vapor at any temperature.

26 No novelty is claimed for this method since the writer found, while preparing this paper, that this very means had been proposed by James Apjohn. However, it does not seem to have been taken very seriously by contemporary scientists since it was never properly developed. Moreover, the details of his method were such as to make it worthless.

27 This method consists, first, in bringing a continuous supply of air close to saturation, where its moisture content can be determined with great accuracy by means of a wet- and dry-bulb thermometer and applying the rational psychrometric formula [3] assuming an approximate value for the specific heat; second, in heating this current of air of known moisture content to any desired amount and taking the wet-and dry-bulb readings as in experiment No. 2, Appendix No. 1. By applying the rational psychrometric formula derived in Pars. 45–53, we have

or $$C_{pa} = \frac{r'(W'-W) - C_{ps}W(t-t')}{(t-t')} \quad \ldots\ldots\ldots\ldots [3]$$

where

C_{pa} = mean specific heat of air of constant pressure between temperatures t and t'

C_{ps} = mean specific heat of water vapor between temperatures t and t'

t = temperature of the dry bulb

t' = temperature of the wet bulb

r' = latent heat of water vapor corresponding to t'

W = weight of water vapor actually contained in 1 lb. of dry air; i.e., it is the ratio of the weight of water vapor to the weight of air in the mixture

W' = weight of water vapor contained in 1 lb. of dry air at saturation at temperature t'

$$W' = \frac{Se'}{P-e'} \quad \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

where

S = specific weight of water vapor

P = barometric pressure

e' = vapor pressure at t' in inches of mercury

DERIVATION OF A RATIONAL PSYCHROMETRIC FORMULA

28 As already pointed out, it is possible to derive a rational psychrometric formula based on the fundamental principles, A, B and C.

29 In considering the interchanges of heat occurring in psychrometric phenomena, it is essential to consider primarily the relative weights of dry air and of water vapor rather than the usual density-temperature relationship; that is, it is necessary to express moisture content as weight of water vapor per pound of pure air, rather than as weight of water vapor per cubic foot of space. Moreover, this relationship is much more adaptable to all of the usual calculations in air conditioning and in meteorology. The author, accordingly, has constructed all his formulae and psychrometric charts upon this basis. In the deduction of the formulae and in the construction of the accompanying charts, the following fundamental data were employed:

a Standard barometric pressure = 29.92 in. mercury = 14.6963 lb. per sq. in. = 2116.3 lb. per sq. ft.

b Absolute temperature $= t + 459.64$ deg. fahr.

c B.t.u.

$$= \frac{\text{heat required to raise 1 lb. of water from 32° to 212°}}{180}$$

d Mechanical equivalent of heat = 777.52 ft-lb.

e Specific volume of air = weight of 1 cu. ft. of pure air at 32 deg. fahr. and 29.92 in. barometric pressure = 0.080728 lb. per cu. ft. Therefore $\frac{pv}{T} = 53.35$.

f Instantaneous specific heat of air[1]

$$C_{pa} = 0.24112 + 0.000009\ t \text{ deg. fahr.}$$

g Vapor pressure, Holborn and Henning's modification of the Theisen formula $(t + 495.6) \log \frac{p}{14.70} = 5.409\ (t - 212) - 3.71 \times 10^{-10}[(689 - t)^4 - (477)^4]$, as calculated in tables of Marks and Davis (1909)

h Specific volume of steam as calculated in steam tables of Marks and Davis[2] (1909)

i Latent heat of water vapor[3]

$r = 141.124\ (689 - t)^{0.31249}$ (t = deg. fahr.)

$r = 1091.16 - 0.56\ t$.

(approximately between 40 and 150 deg. fahr.)

j Instantaneous specific heat of water vapor (approximately)

$C_{ps} = 0.4423 + 0.00018\ t$ deg. fahr.

k Specific weight of water vapor at saturation for any pressure and temperature

$$S = \frac{\text{specific volume of air}}{\text{specific volume of steam}}$$

30 With respect to the reliability of these data, those on the latent heat of steam may be accepted as absolute within 0.1 per cent, since the agreement of recent investigators seems to have established the present values beyond question. Strange to say, however, the specific heat of air, as already pointed out, has not been established with accuracy within 2 per cent. Regnault gives it as a constant $C_p = 0.2375$, and this value has generally been accepted. However, Holborn and Henning, whose valuable determinations in steam are

[1] W. F. G. Swann, Phil. Trans. Royal Soc., series A, vol. 210, pp. 199–238, 1909.

[2] Harvey N. Davis, Trans. Am.Soc.M.E., vol. 30, p. 750, 1908.

[3] C. H. Peabody, Trans. Am.Soc.M.E., vol. 31, p. 334, 1909.

well known, have demonstrated it to be a variable. For nitrogen they give a value $C_{(o-t)} = 0.2350 + 0.000019\ t$ (t in deg. cent.), a straight-line relationship, although for superheated vapors they find equations of a higher degree. If their values were accepted at atmospheric temperatures we would have a specific heat for air considerably lower than that given by Regnault, while psychrometric evidence seems to indicate that it would be considerably higher at such temperatures. The most reliable of recent determinations would seem to be that of W. F. G. Swann.[1] In his paper he points out a defect in the method of Regnault which would account for the latter's value being too low. The values given by Swann have therefore been adopted in this paper; although they appear still to require confirmation, since there would seem to be considerable opportunity for error in the method of air measurement used in his experiments.

31 The equation given for the specific heat of steam at low temperatures seems to agree fairly well with modern experimental data. Extreme accuracy is not pretended, nor is this essential at lower temperatures, since under 150 deg. the total heat value of the air is affected only 2 per cent at most by the specific heat of the water vapor. The values given by this equation, however, are undoubtedly more nearly correct at lower temperatures than the usual value, $C = 0.48$.

32 The psychrometric charts, Figs. 1 and 2, are constructed accurately from the foregoing data. Fig. 2 exhibits all psychrometric relationships, between the temperatures of 20 deg. and 350 deg. and saturation temperatures up to 143 deg. Fig. 1 gives the same values between temperatures 20 deg. and 110 deg. and saturation temperatures to 95 deg. These charts are here shown to a greatly reduced scale. In its original form, Fig. 1 permits the reading of both the wet- and dry-bulb temperatures to an accuracy of 0.1 deg. and of the moisture weight per pound of air to 0.2 grains. All calculations have been made with accuracy to five significant figures by means of a Thatcher slide rule.

SATURATION CURVE

33 The saturation curve, Fig. 2, expressed in grains of moisture per pound of air, was computed from the formula

[1] Phil. Trans. Royal. Soc., series A, vol. 210, pp. 199–238, 1909.

$$G = \frac{5284\ (t + 459.64)\ D_s}{P - e} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

where

G = grains of moisture per lb. of pure air at saturation
t = temperature of saturation in deg. fahr.
$t + 459.64$ = absolute temperature
D_s = density in lb. per cu. ft. of saturated water vapor at temperature t
= reciprocal of specific volume of steam
$P = 29.92$ = assumed standard of barometric pressure in in. of mercury
e = vapor pressure of saturated water vapor in inches of mercury
5284 = constant of the equation

The derivation of equation [5] is given in Appendix No. 2.

34 The specific weight of saturated water vapor is not constant, but varies with the temperature of saturation and may be calculated from equation [1]. The theoretical value, computed from its molecular weight, assuming it to be a perfect gas, is 0.6221. This is the assumption made in the computation of the psychrometric tables published by the United States Weather Bureau, which are inaccurate, therefore, in proportion to the variation of S from this theoretical value. The actual values for the specific weight of water vapor at various saturation temperatures, computed from equation [37] are given in Table 1.

35 The specific weight of water vapor may also be given independent of the density as

$$S = 0.6221 + 0.001815\ \sqrt{e} + 0.0000051\ \sqrt{e^3}$$

Hence

$$G = \frac{7000\ (0.6221e + 0.00182\ \sqrt{e^3} + 0.0000051\ \sqrt{e^5})}{P - e}$$

At atmospheric temperature the term $0.0000051\ \sqrt{e^5}$ is negligible. Hence

$$G = \frac{7000\ (0.6221e + 0.00182\ \sqrt{e^3})}{P - e}$$

CONSTRUCTIONS OF ADIABATIC SATURATION LINES

36 As shown in Par. 22 any adiabatic change involving moisture content and temperature of air may be expressed by equating the change in total specific heat to the corresponding change in latent

heat. It may also be expressed by equating the total heat contained in the air in any state to its total heat when in the state of adiabatic saturation. By either of these two methods, given in Appendix

TABLE 1 SPECIFIC WEIGHT OF STEAM

$$S = \frac{53.35\ (t + 459.6)\ D_s}{144 \times 0.4908\ e}$$

t	e	D_s	S	S Corrected	D_s Corrected	Grains Moisture per cu. ft.
40	0.2477	0.000410	0.6245	0.6228	0.0004089	2.8623
50	0.3625	0.000587	0.6230	0.6231	0.0005871	4.1697
60	0.5220	0.000828	0.6221	0.6233	0.0006296	4.4072
70	0.7390	0.001148	0.6212	0.6236	0.0011524	8.0668
80	1.0290	0.001570	0.6217	0.6239	0.0015755	11.0285
90	1.4170	0.002130	0.6237	0.6242	0.0021320	14.9240
100	1.9260	0.002851	0.62525	0.6246	0.0028482	19.9374
110	2.5890	0.003766	0.6255	0.6250	0.0037630	26.3410
120	3.4380	0.004924	0.62665	0.6254	0.0049140	34.3980
130	4.5200	0.006370	0.6273	0.6260	0.0063570	44.4990
140	5.8300	0.008140	0.6266	0.6266	0.0081400	56.9800
150	7.5700	0.010320	0.6280	0.6273	0.0103100	72.1700
160	9.6500	0.012960	0.6282	0.6280	0.012956	90.6920
170	12.200	0.016140	0.6288	0.6288	0.016140	112.9800
180	15.290	0.019940	0.6296	0.6296	0.019940	139.5800
190	19.020	0.024440	0.6301	0.6307	0.024465	171.2550
200	23.47	0.029760	0.6313	0.6316	0.029780	208.4600

No. 3, the relations of formula [6] are established, i.e.

$$r'(W' - W) = C_{pa}(t - t') + C_{ps} W (t - t') \ldots\ldots\ldots [6]$$

in which

$(t - t')$ = the true wet-bulb depression

$(W' - W)$ = the moisture absorbed per lb. of pure air when it is adiabatically saturated from an initial dry-bulb temperature t_o and an initial moisture content W

C_{pa} = mean specific heat of air at constant pressure between temperature t and t'

C_{ps} = specific heat of steam at constant pressure between t and t'

r' = latent heat of evaporation at wet-bulb temperature t'

Knowing any two of the three important values of t, t' or W, we may solve for the third or for any other required relation.

37 *Determination of Weight of Moisture in 1 Lb. of Pure Air, having a Dry-Bulb Temperature t and Wet-Bulb Temperature t'.* To determine the equation of the adiabatic line corresponding to a given saturation, or true wet-bulb temperature t', and dry-bulb temperature t, we have from equation [6]

$$[r' + C_{ps}(t - t')]\, W = r'W' - C_{pa}(t - t') \quad \ldots\ldots [7]$$

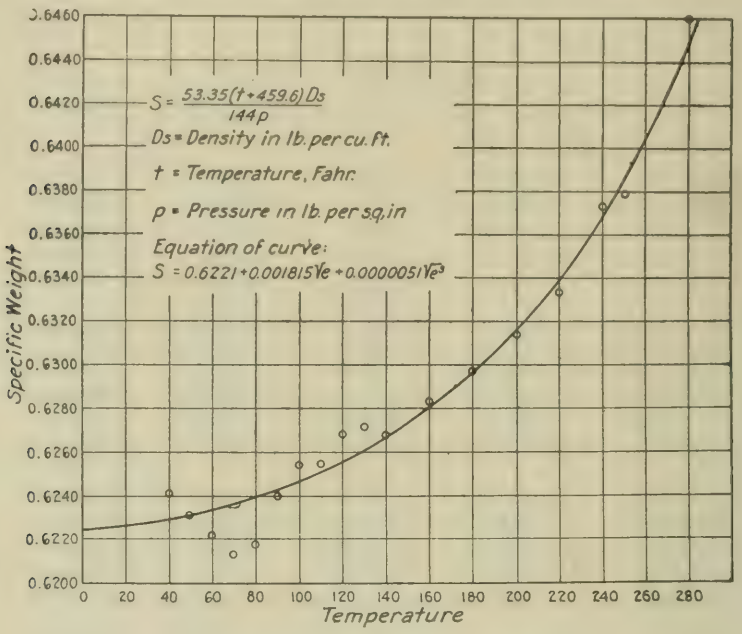


FIG. 4 SPECIFIC WEIGHT OF STEAM

$$W = \frac{r'W' - C_{pa}(t - t')}{r' + C_{ps}(t - t')} \quad \ldots\ldots [8]$$

38 The diagonal adiabatic lines in the charts, Figs. 1 and 2, representing saturation or wet-bulb temperatures, are calculated from this formula. It should be observed that they would be perfectly straight if it were not for the element $C_{ps}(t-t')$, which produces a slight curvature, becoming more pronounced at higher saturation temperatures. The dewpoint t_1 corresponds to W on the saturation curve. The slope of these lines, neglecting $C_{ps}(t-t')$, is $\frac{dW}{dt} = -\frac{C_{pa}}{r'}$. This will always give the intercept t for $W = 0$.

39 *Wet-Bulb Depression and Cooling Effect.* The wet-bulb depression or cooling effect obtained by having t and W known is

$$t - t' = \frac{r'(W' - W)}{C_{pa} + C_{ps}W} \quad \ldots\ldots [9]$$

and

$$t = t' + \frac{r'(W' - W)}{C_{pa} + C_p W} \quad [10]$$

40 Having t and W known, t' cannot be calculated except by relating W to t by an empirical equation. By referring to the psychrometric charts, Figs. 1 and 2, constructed chiefly for that purpose, t' is conveniently determined. The cooling effect $(t - t')$, to be obtained by saturating air of known temperature and moisture content, is likewise obtained from the chart.

41 *Moisture-Absorbing Capacity of Air.* For determining the moisture-absorbing capacity, or moisture deficit of air, having a known temperature and moisture content, we have per pound of pure air

$$(W' - W) = \frac{(C_{pa} + C_{ps} W)(t - t')}{r'} \quad [11]$$

42 *Per Cent Adiabatic Saturation.* The per cent of adiabatic saturation is

$$\frac{W}{W'} = \frac{r' W' - C_{pa}(t - t')}{r' W' - C_{ps}(t - t') W'} \quad [12]$$

or if we neglect $C_{ps}(t - t') W'$

$$\frac{W}{W'} = 1 - C_{pa} \frac{(t - t')}{r' W'} \quad [13]$$

43 *Specific Heat of Air.* The equation for the experimental determination of the specific heat of air by the evaporative method is

$$C_{pa} = \frac{r'(W' - W) - C_{ps}(t - t') W}{(t - t')} \quad [14]$$

44 For engineering purposes however, it is preferable to determine any unknown value directly from the psychrometric charts, which afford to a great degree of accuracy a simple graphic solution of any problem of psychrometry.

DERIVATION OF THE RATIONAL PSYCHROMETRIC FORMULA FOR VAPOR PRESSURE

45 The present empirical psychrometric formula in use by the United States Weather Bureau was first deduced by Professor Ferrel,[1] while the constants of the formula were deduced from a series

[1] Annual Report, Ch. Signal Officer, 1886, Appendix 24, pp. 233–259.

of experiments by Professor Marvin and Prof. H. A. Hazen. The error in Broch's and Regnault's values for vapor pressures which they adopted alone would require its revision. Moreover, the oppor-

TABLE 2 COMPARISON OF NEWLY DETERMINED PSYCHROMETRIC VALUES WITH OLD UNITED STATES WEATHER BUREAU VALUES

Water Vapor Contained in 1 Lb. of Pure Air for Wet Bulb of 60 Deg. Fahr. and Dry Bulb of t from U. S. Weather Bureau Formulæ.						Water Vapor Actually Contained in 1 Lb. of Pure Air for Wet Bulb of 60 Deg. Fahr.				
t	$t-t'$	e	$29.92-e$	Grain per Lb. Air	Per Cent Error	t	$t-t'$	C_{ps}	C_{pa}	Grain per Lb. Air
70	10	0.4053	29.515	59.8	0.0205	70	10	0.4540	0.24170	61.06
75	15	0.3495	29.570	51.5	0.0280	75	15	0.4544	0.24172	52.98
80	20	0.2936	29.626	43.1	0.0388	80	20	0.4549	0.24175	44.84
85	25	0.2380	29.682	34.9	0.0546	85	25	0.4553	0.24177	36.915
90	30	0.1820	29.738	26.6	0.0808	90	30	0.4558	0.24179	28.934
95	35	0.1260	29.794	18.4	0.1236	95	35	0.4562	0.24181	20.995
100	40	0.0700	29.850	10.2	0.2200	100	40	0.4567	0.24184	13.082
105	45	0.0145	29.905	2.11	0.5960	105	45	0.4571	0.24187	5.220

$$e = e' - 0.000367\,P\,(t-t')\left(1+\frac{t'-32}{1571}\right)$$

$$G = \frac{e \times 0.6221 \times 7000}{29.92 - e}$$

t = dry-bulb temperature, deg. fahr.

C_{pa} = specific heat of air

C_{ps} = specific heat of vapor

G' = 77.32 grains per lb.

t' = 60 deg. fahr.

r' = 1057.8

$C_{ps} = 0.4423 + 0.00018\,t_m$

$C_{pa} = 0.24112 + 0.000009\,t_m$

$$G = \frac{G'r' - 7000\,C_{pa}\,(t-t')}{r' + C_{ps}\,(t-t')}$$

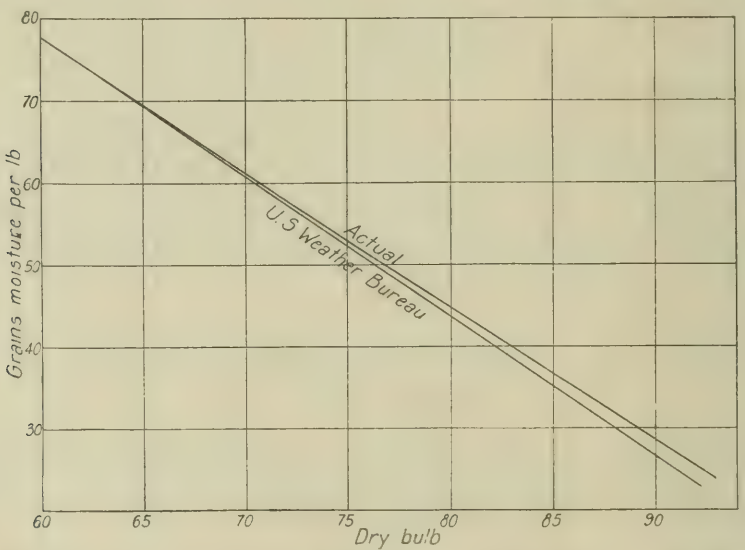


FIG. 5 COMPARISON OF ACTUAL MOISTURE CONTENT OF AIR AT 60 DEG. WET BULB WITH THAT GIVEN IN U. S. WEATHER BUREAU TABLES

tunity for error in their methods has already been pointed out. However, considering the difficulties of their experimental method and the correction necessary for radiation in the wet-bulb reading, the results obtained are remarkable for consistency and accuracy. There are errors in the form of the equation, however, as well as in the constants employed, which make its inaccuracy more pronounced at lower humidities and at the higher temperature. At very high temperatures used in mechanical drying it is entirely inoperative.

46 The need of an accurate rational psychrometric formula for vapor pressures using modern data is therefore apparent. The required values could be obtained indirectly from the formula already given, but computation is facilitated by another derivation giving directly the vapor pressure e.

47 In equation [8]

$$W = \frac{r' W' - C_{pa}(t - t')}{r' + C_{ps}(t - t')}$$

let t be the dewpoint corresponding to W, and e the vapor pressure corresponding to W in inches of mercury. Referring to equation [4]

$$W = \frac{Se}{P - e}$$

By substitution

$$\frac{S_1 e}{P - e} = \frac{r'\left(\frac{S'e'}{P - e'}\right) - C_{pa}(t - t')}{r' + C_{ps}(t - t')} \quad \ldots\ldots\ldots\ldots [15]$$

Solving for e

$$e = P\left\{\frac{\left[\frac{S'e'r' - (P - e')\,C_{pa}(t - t')}{(P - e')\,[r' + C_{ps}(t - t')}\right]}{\left\{\frac{S_1(P - e')\,[r' + C_{ps}(t - t')] + [S'e'r' - (P - e')\,C_{pa}(t - t')]}{(P - e')\,[r' + C_{ps}(t - t')]}\right\}}\right\} \quad [16]$$

Assuming $S'e' = S_1 e_1$, this simplifies to

$$e = P\left[\frac{S'e'r' - (P - e')\,C_{pa}(t - t')}{S_1 P r' + (S_1 C_{ps} - C_{pa})\,(P - e')\,(t - t')}\right] \quad \ldots\ldots [17]$$

At 100 deg.

$$S_1 C_{ps} - C_{pa} = [(0.623 \times 0.46) - 0.242] = 0.033 \quad \ldots\ldots [18]$$

Therefore the value $-e\,(S_1 C_{ps} - C_{pa})\,(t - t')$ is ordinarily negligible. Hence we have

$$e = \frac{S'e'r' - C_{pa}(P-e')\,(t-t')}{Sr' + (S_1 C_{ps} - C_{pa})\,(t-t')} \quad \ldots\ldots\ldots\ldots [19]$$

48 By comparison it will be found that the difference between $S'r'$ and $Sr' + (S_1\,C_{ps} - C_{pa})\,(t-t')$ is negligible. Hence it may be assumed

$$e = e' - \frac{C_{pa}\,(P-e')\,(t-t')}{S'r'} \quad \ldots\ldots\ldots\ldots [20]$$

For r' may be substituted the approximate value $r' = 1091.6 - 0.56\,t'$ and for S' the value $S' = 0.6215 + 0.000034\,t'$ (approximately). Hence

$$e = e' - \frac{C_{pa}\,(P-e')\,(t-t')}{(0.6215 + 0.000034t')\,(1091.6 - 0.56t')} \quad \ldots\ldots [21]$$

$$e = e' - \frac{C_{pa}\,(P-e')\,(t-t')}{678.4 - 0.3011\,t'} \quad \ldots\ldots\ldots\ldots [22]$$

and if Swann's value of $0.24112 + 0.000009\,t$ is accepted for the value of C_{pa} as in the charts

$$e = e' - \frac{(P-e')\,(t-t')}{2803 - 1.329\,t'} \quad \ldots\ldots\ldots\ldots [23]$$

This equation has been carefully tested by comparing with values of the psychrometric coefficient as determined by means of the charts as shown in Table 2 and Fig. 6. This shows that it is permissible to use $\frac{1}{2800 - 1.3\,t'}$ as the coefficient instead of $\frac{1}{2803 - 1.329\,t'}$, giving

$$e = e' - \frac{(P-e')(t-t')}{2800 - 1.3\,t'} \quad \ldots\ldots\ldots\ldots [24]$$

This formula will give values of e for all wet- and dry-bulb temperatures and all barometric pressures with an error of less than 0.5 per cent, assuming the chosen value of C_{pa} to be correct.

49 This equation should be used where the true wet-bulb temperature is obtained as in the aspiration psychrometer. With the sling psychrometer a correction must be made for the error in depression due to radiation and stem correction. By referring to Fig. 7 showing the per cent of radiation error, it is seen that this is inversely proportional to the velocity. It is also, of course, greatly affected by the conditions of exposure, i.e., whether it is surrounded on all four sides

by bodies at the temperature of the dry bulb, or only partly by bodies of that or a different temperature. The effect of radiation outside of an enclosure may be assumed to be approximately one-half of that within an enclosure.

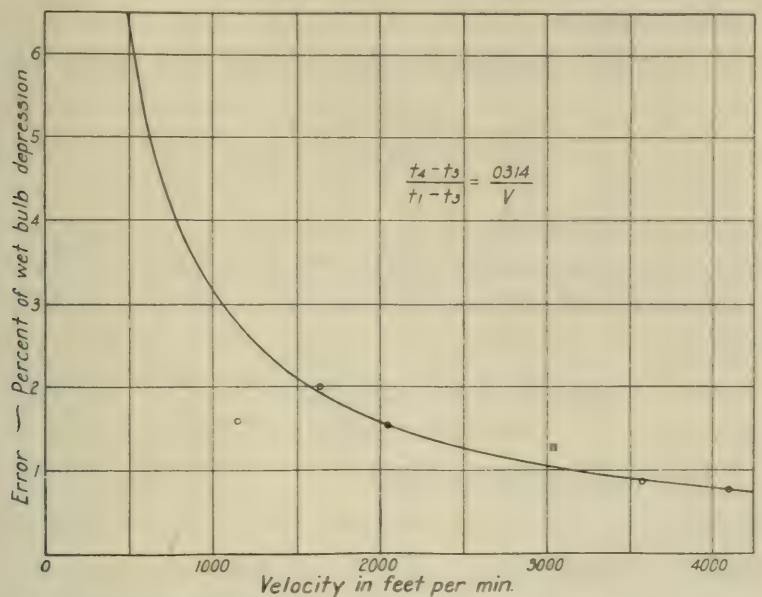


FIG. 6 RADIATION ERROR IN WET BULB OF SLING PSYCHROMETER

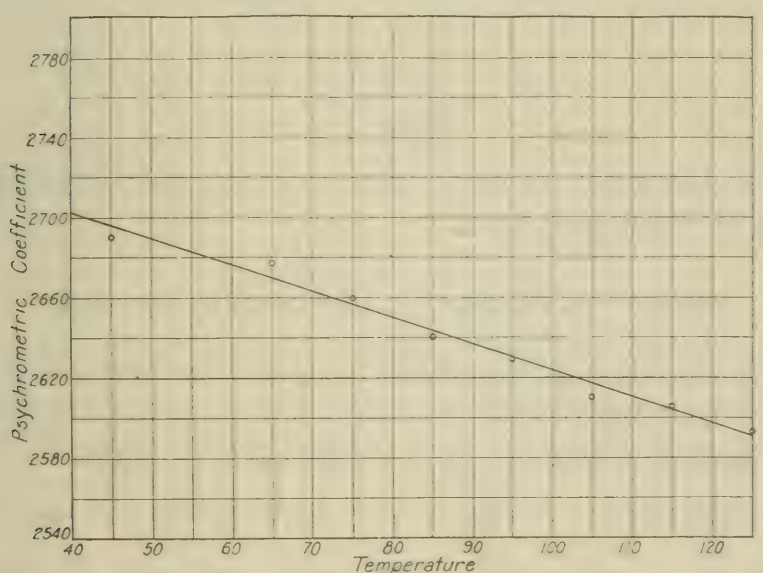


FIG. 7 PSYCHROMETRIC COEFFICIENT

50 A sling psychrometer 15 in. in length is ordinarliy revolved between 150 and 225 r.p.m. giving a velocity between 1200 and 1800 ft. per min. This will give, according to Fig. 7, a radiation error of 2.6 to 1.75 per cent within an enclosure, and 1.3 to 0.9 per cent without an enclosure. Hence an average radiation error of 1.6 per cent

of the wet-bulb depression may be arbitrarily assumed. The wet-bulb depression given by the sling psychrometer may be corrected by this amount to give the true depression, which may be used in the foregoing psychrometric formula, or the formula itself may be modified to allow for this error.

51 If this formula is corrected for 1.6 per cent radiation error

$$e = e' - \frac{(P - e')\,(t - t')}{2755 - 1.28\,t'} \quad \ldots\ldots\ldots\ldots\ldots\ldots [25]$$

for the sling psychrometer.

52 Using the true wet-bulb depression in formula [24], letting e_2 be the vapor pressure corresponding to saturation at the dry-bulb temperature t

$$R = \frac{e}{e_2} = \frac{e'}{e_2} - \frac{(P - e')\,(t - t')}{(2800 - 1.3\,t)\,e_2} \quad \ldots\ldots\ldots\ldots [26]$$

for the per cent of relative humidity.

53 Let W be the grains of moisture per cu. ft. at any vapor pressure e, and W_2, grains per cu. ft. at e_2; then

$$W = (RW_2) = \frac{W_2}{e_2}\left[e' - \frac{(P - e')\,(t - t')}{2800 - 1.3\,t'}\right] \quad \ldots\ldots\ldots [27]$$

also

$$W = \left(\frac{460 + t'}{460 + t}\right) W_1 \quad \ldots\ldots\ldots\ldots\ldots\ldots [28]$$

where W = the grains of moisture per cu. ft., corresponding to the dewpoint at vapor pressure e.

EFFECT OF CHANGE IN BAROMETRIC PRESSURE

54 Suppose that air in which the vapor pressure is e_0 is compressed from a barometric pressure P_0 to a barometric pressure P, then the partial pressure of both the air and the vapor are increased proportionally and $e = \frac{e_0 P}{P_0}$. The temperature corresponding to saturation at e is the temperature of the dewpoint at pressure P.

55 The per cent of isothermal saturation becomes

$$R = \left(\frac{e}{e_2}\right) = \frac{e_0}{e_2}\,\frac{P}{P_0} = \frac{R_0 P}{P_0} \quad \ldots\ldots\ldots\ldots [29]$$

where e_2 is the saturated vapor pressure corresponding to the dry-bulb temperature t.

TOTAL HEAT CURVE

56 This curve shows the sensible heat in the air above a base temperature of 0 deg. fahr., plus the latent heat contained in the water vapor at saturation, but not including the heat of the liquid. Since the wet-bulb temperature, or adiabatic lines contain all points having the same total heat (neglecting heat of liquid), the curve serves to determine the total heat in the air under any and all conditions represented by the chart. This is of great convenience in calculating refrigeration required to cool and dehumidify air. For example, suppose it is required to find the refrigeration necessary to cool 1 lb. of air containing 98 grains of moisture and having a dry-bulb temperature of 95 deg., to a final temperature of 40 deg. saturated. We find from the chart that the wet-bulb temperature is 75 deg. The total heat corresponding to a saturation temperature of 75 deg. is 37.8 B.t.u., while the total heat at 40 deg. is 15.3 B.t.u. The difference, 22.5 B.t.u. is the refrigeration required per pound of air.

The author wishes to acknowledge his indebtedness to his assistants, Mr. Theodore A. Weager and Mr. Frank L. Busey, for the actual work of computation and the construction of the diagrams.

APPENDIX No. 1

EXPERIMENTAL DEMONSTRATION OF THE FOUR PSYCHROMETRIC PRINCIPLES GIVEN IN PARS. 22 AND 23

57 While the four psychrometric principles might be all logically surmised, experimental demonstration is desirable. A calorimetric method was devised

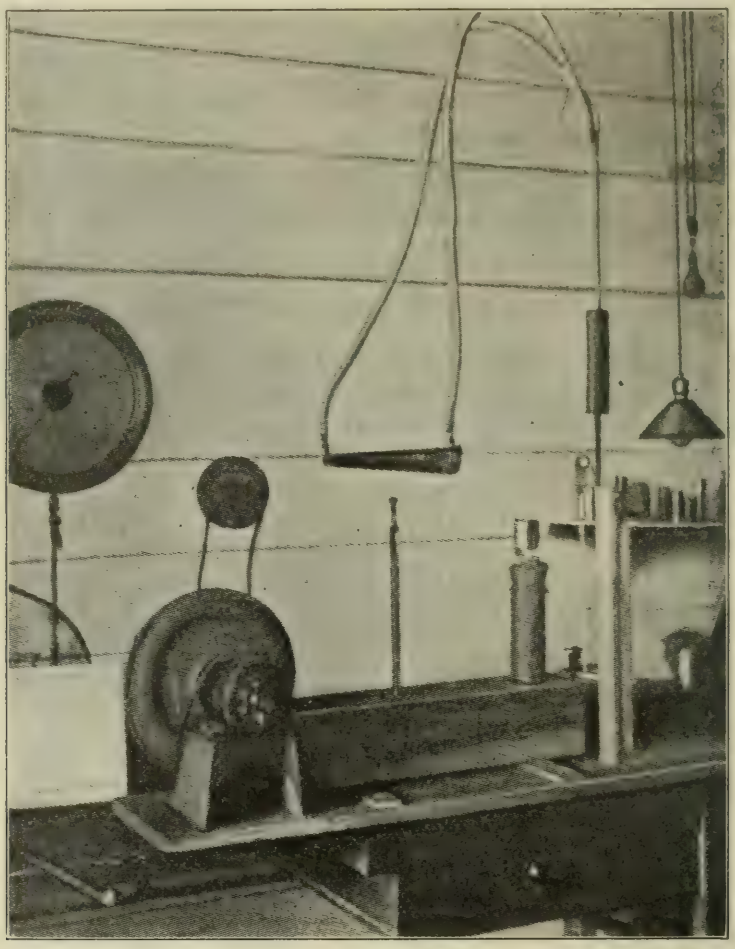


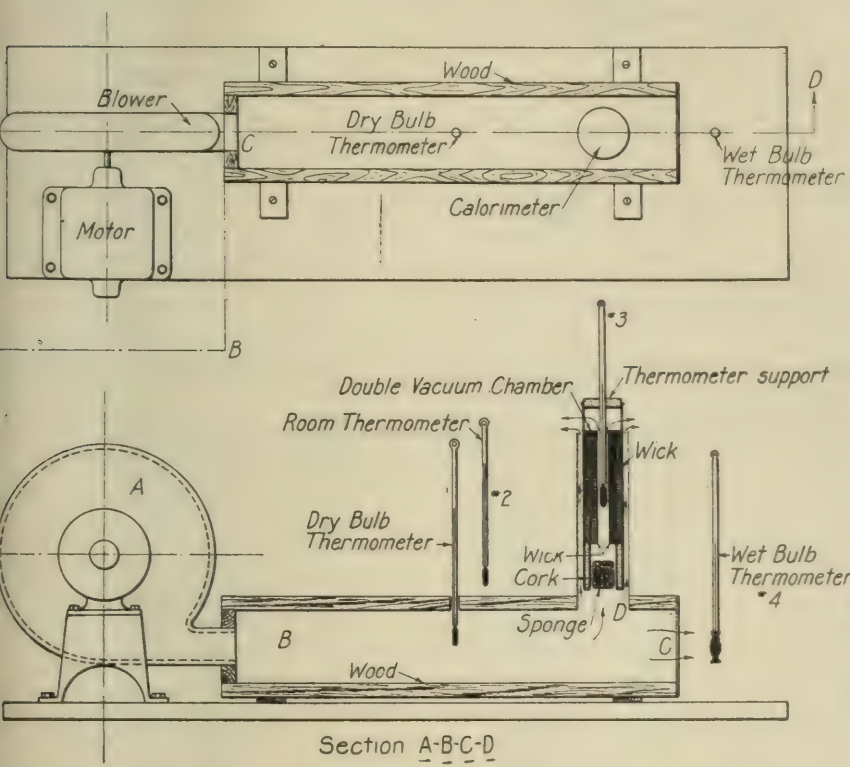
FIG. 8 GENERAL VIEW OF APPARATUS

by the author for this purpose and also to determine the probable error of the indications of the wet bulb in the sling psychrometer due to radiation.

58 The apparatus used is shown in Figs. 8, 9, and 10. Air was supplied by fan *A* at slight pressure to the wooden air duct *B*, from which it escaped through the orifice *C*, and through the tube *D*, in which the wet-bulb calo-

rimeter, Figs. 8 and 9, was used in different experiments. A differential draft gage and pitot tube indicated the velocity of the air through the orifice and over the wet-bulb thermometer No. 4.

59 This velocity could be varied any desired amount between 1000 and 4000 ft. per min. by adjustment of the motor rheostat. It was found that the static pressure in the box agreed substantially with the velocity head at the thermometer bulb so that no further measurements of the former were recorded. Thermometer No. 1 indicated the dry-bulb temperature of the air in the box, thermometer No. 2, the dry-bulb temperature of the air outside, and thermometer No. 3, the calorimeter temperature. Thermometers Nos. 3 and 4 were calorimeter thermometers especially constructed for this test by the Taylor



. 9 PLAN AND ELEVATION OF APPARATUS

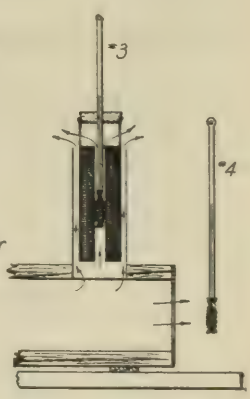


FIG. 10 DETAIL OF CALORIMETER

Instrument Company. They were in the fahrenheit scale, graduated to tenths of a degree, and calibrated to $\frac{1}{40}$ deg. They were also carefully compared.

60 *Experiment No. 1.* This test was made in order to determine the effect of an air blast of known intensity upon the readings of thermometers No. 3 and No. 4. The need of the determination was evident as the velocities were not necessarily the same upon the two bulbs nor in the same relative direction. Moreover, it was evident that a portion of the heat of the air was converted into mechanical energy of the air current; also that a portion of this, at least, was re-converted into heat by impact on the bulb. This temperature error, if any, would be proportional to the velocity head; therefore a maximum condition of 1-in. velocity head and static pressure were taken. Both thermometers and the calorimeter were perfectly dry. The apparatus was run under constant

conditions for 1 hour previous to the test. Consecutive readings were taken of marked uniformity, and the average results are given in Table 3.

61 The actual calculated drop in temperature due to 1-in. air blast under the above conditions is

$$D = (82.7 + 459.6)\left[1 - \left(\frac{397}{398}\right)^{0.29}\right] = 0.38 \text{ deg.} + \ldots\ldots\ldots [30]$$

62 It will be noted that thermometer No. 3 read 0.047 lower than No. 4 at 1 in pressure. However, at a wet-bulb temperature of 70 deg., 1 deg. in the temper-

TABLE 3 EFFECT OF AIR BLAST OF KNOWN INTENSITY ON READINGS OF THERMOMETERS NO. 3 AND NO. 4

Thermometers				Temperature Differences		
No. 2	No. 1	No. 4	No. 3	1 and 3	1 and 4	3 and 4
.....	82.737	.....	82.653	0.084	.....	.....
.....	.....	82.566	82.613	.....	.....	0.047
.....	.....	.....	.....	.....	0.131	.....

Barometer pressure = 29.3 in. mercury = 397 in. water.

TABLE 4 DIFFERENCES IN THERMOMETER READINGS 1 AND 3 WITH BOTH BULBS DRY

Static Pressure in Box, 1.0 in.; Velocity Pressure at Thermometer, 1.0 in.

No. 1 Dry Bulb	No. 3 Calorimeter	Difference
82.7	82.625	0.075
0.7	0.600	0.100
0.65	0.600	0.050
0.7	0.650	0.050
0.8	0.675	0.125
0.8	0.675	0.125
0.85	0.750	0.100
0.7	0.650	0.050
Average Difference.....		0.084

ature of the dry-bulb produces only 0.3 deg. increase in wet-bulb temperature; i.e., $\frac{dt'}{dt} = 0.30$. Therefore at 1 in. pressure the error would be 0.014 and at $\frac{1}{4}$ in. pressure 0.0035. Hence, in any case, the correction would be negligible. The temperature increase E produced by an air blast equivalent to p in. of water may be expressed by the equation

$$E = \left(\frac{0.38 - 0.13}{1}\right) p = 0.25\ p \ldots\ldots\ldots\ldots\ldots [31]$$

or $\frac{0.25}{0.38} = 66$ per cent of the theoretical temperature.

63 *Experiment No. 2.* This was for the purpose of determining approximately the per cent of error due to radiation and stem correction in the depression of the wet bulb of the sling psychrometer, and was accomplished by comparing wet-bulb thermometer No. 4, with wet-bulb thermometer No. 3, which was protected from practically all radiation by surrounding it with insulated wet surfaces at precisely the same temperature, and by protecting the stem with a wet cloth. This arrangement is shown in Fig. 10, and may be termed a wet-bulb or evaporation calorimeter. The protection for the wet bulb consisted in an annular vacuum tube, having the exterior surface covered with a wet cloth, and the interior with a tube of wet blotting paper. This was

TABLE 5 DIFFERENCES IN THERMOMETER READINGS 3 AND 4 WITH BOTH BULBS DRY

No. 3 Calorimeter	No. 4 Thermometer	Difference	No. 3 Calorimeter	No. 4 Thermometer	Difference
82.75	82.725	0.025	82.55	82.500	0.050
82.72	82.650	0.070	82.525	82.500	0.025
82.70	82.700	0.000	82.70	82.700	0.000
82.80	82.750	0.050	82.70	82.675	0.025
82.80	82.750	0.050	82.70	82.600	0.100
82.80	82.750	0.050	82.60	82.500	0.100
82.75	82.650	0.100	82.50	82.400	0.100
82.70	82.600	0.100	82.40	82.300	0.100
82.65	82.600	0.050	82.45	82.400	0.050
82.65	82.575	0.075	82.50	82.475	0.025
82.60	82.525	0.075	82.55	82.500	0.050
82.60	82.525	0.075	82.50	82.500	0.000
82.50	82.450	0.050	82.55	82.550	0.000
82.50	82.475	0.025	82.60	82.600	0.000
82.60	82.575	0.025	82.65	82.610	0.040
82.60	82.600	0.000	82.60	82.600	0.000
82.60	82.500	0.100	82.55	82.430	0.120
82.50	82.425	0.075	82.60	82.600	0.000
82.50	82.450	0.050	82.70	82.700	0.000
82.55	82.525	0.025	82.75	82.725	0.025
			Average		0.047

placed in an open tube leading from the air duct so that there is the same circulation of air over the wet surfaces as over the wet-bulb thermometer placed within. Thermometer No. 4 was rotated while being subjected to the blast, so that the condition in the sling psychrometer would be exactly reproduced.

64 The log of these tests shows that the error in the depression of the wet bulb in the sling psychrometer for various velocities is as shown in Table 6 and Fig. 6. This error has been taken as directly proportional to the depression. More accurately it is proportional to the difference of the fourth powers of the respective absolute temperatures, except for the stem correction, However, where the depression is the usual small percentage of the absolute temperature, the error in assuming direct proportionality is insignificant.

65 The sling psychrometer, however, is subject to another error, heretofore seemingly overlooked. As shown in experiment No. 1, there is a rise in temperature due to the impact of the air upon the bulb, which, in the case of the dry bulb is 66 per cent of the theoretical, or 0.25 p, and in the case of the wet bulb, $0.25p\frac{dt'}{dt}$. This, however, would have no effect upon the calculated absolute humidity, but only on the temperature and consequent relative humidity.

66 The type of psychrometer which lends itself to the most accurate determinations is the Aszmann aspiration psychrometer[1] shown in Fig. 11. Here the air is aspirated through two tubes containing the wet and dry-bulb thermometers. The wet-bulb temperature is brought to a minimum by the use of an atomizer.

TABLE 6 AVERAGE RESULTS OF TEST FOR RADIATION ERROR IN WET BULB OF SLING PSYCHROMETER

1	Velocity pressure, in. water.....	0.08	0.16	0.25	0.55	0.75	1.00
2	Velocity, ft. per min.....	1160	1640	2050	3040	3550	4100
3	Room temperature No. 2.....	86.0	82.0	85.1	85.2	87.2	87 25
4	Dry-bulb temperature No. 1....	87.125	83.93	83.88	83.975	88.175	88.59
5	Calorimeter temperature No. 3..	68.935	69.725	71.158	71.740	68.494	68.830
6	Wet-bulb temperature No. 4.....	89.21	70.01	71.353	71.892	68.654	68.965
7	Calorimeter depression (difference between items 4 and 5)..	18.19	14.205	12.722	12.235	19.681	19.760
8	Difference between wet-bulb and calorimeter temperature.....	0.295	0.285	0.195	0.152	0.160	0.135
9	Ratio of wet bulb minus calorimeter temperature to calorimeter depression (item 8 ÷ 7).	0.016	0.020	0.015	0.012	0.008	0.007
10	Item 9 corrected for difference in impact.....	0.01606	0.02018	0.01527	0.01239	0.00853	0.00771

Temperature difference due to impact = 0.3 × 0.047 × velocity pressure.

$\frac{dt'}{dt}$ = 0.3 at t' = 69 to 70 deg. fahr.

0.047 = temperature difference (thermometer No. 3-thermometer No. 4) with dry-bulb thermometers due to difference in impact at 1 in. velocity pressure.

This serves also to moisten the inner surface of the enveloping tube, thus cooling it and preventing radiation. In this type of psychrometer it should be noted that the impact of the air upon the thermometer bulbs largely neutralizes the reduction in pressure, producing the velocity as demonstrated in experiment No. 1.

67 *Experiment No. 3.* The purpose of this experiment was to demonstrate principles *C* and *D* in Pars. 22 and 23. A modified form of the evaporation calorimeter was used. The air was first passed through two layers of moistened sponge, bringing it very close to true adiabatic saturation. Its temperature was then taken alternately with a wet-bulb and a dry-bulb thermometer, and simultaneous readings were taken with thermometer No. 4, the error of which had been established. It was found very difficult to

[1] For full description see Zeitschrift für Instrumentenkunde, January 1892.

obtain consistent readings to the degree of accuracy desired, on account of the extreme lag of temperature in the calorimeter thus constructed. On this account it was found necessary to maintain the temperature constant at No. 4 by continuous hand regulation at the fan inlet and continuous observations of thermometer No. 4. It was found possible in this way to prevent a variation of more than 0.05 deg. Results showed the temperature in the calorimeter with a wet bulb to be a little lower than No. 3 when a dry bulb was used, owing to slightly imperfect saturation. This test, therefore, did not agree exactly with the results of experiment No. 2. It appeared to be possible, however, for the water on the wet bulb in experiment No. 2 to be cooled to a lower temperature than that of adiabatic saturation, and

TABLE 7 COMPARISON OF WET-BULB TEMPERATURE WITH SATURATION TEMPERATURE WHEN PASSING AIR THROUGH WET SPONGE IN CALORIMETER, PRESSURE, 0.25 IN.

No. 1 Dry-bulb Temperature	No. 2 Wet-Bulb Temperature	No. 3 Calorimeter Temperature	Difference
81.75	61.90	61.80	0.10
81.75	61.95	61.85	0.10
81.75	62.00	61.85	0.15
81.75	62.05	61.90	0.15
81.75	62.00	61.90	0.10
81.75	62.00	61.90	0.10
81.75	62.65	62.65	0.00
81.75	62.60	62.65	0.05
81.75	62.65	62.65	0.00
81.75	62.65	62.625	0.025
81.75	62.65	62.65	0.00
81.75	62.60	62.625	0.025
81.75	62.65	62.65	0.00
81.75	62.60	62.625	0.025
81.75	62.65	62.65	0.00
81.75	62.60	62.65	0.05
81.75	62.65	62.625	0.025
81.75	62.70	63.65	0.05

it is necessary, therefore, to attribute this slight discrepancy to some source of error in the temperature of the air in experiment No. 3. Three explanations are possible: (*a*) the air being thoroughly saturated before entering the tube of the calorimeter, its temperature would easily be increased with any slight radiation due to imperfect insulation, especially since air delivery was greatly reduced by the resistance of the sponge; (*b*) at the time the readings were taken the outside was always beginning to get dry, due to the very long time required to bring the temperature of the calorimeter to a minimum, during which the cloth on the calorimeter would begin to dry and require moistening, resulting in a momentarily increase of temperature; (*c*) the possibility of some parts of the sponge becoming dry and conducting a slight amount of heat to the wet portions.

68 The agreement of these tests, however, is quite sufficient to warrant fully the acceptance of the fundamental principles previously stated. It is also made evident that the reading of the wet-bulb thermometer properly protected from radiation as in experiment No. 2 is a most practicable and accurate method of determining the temperature of adiabatic saturation.

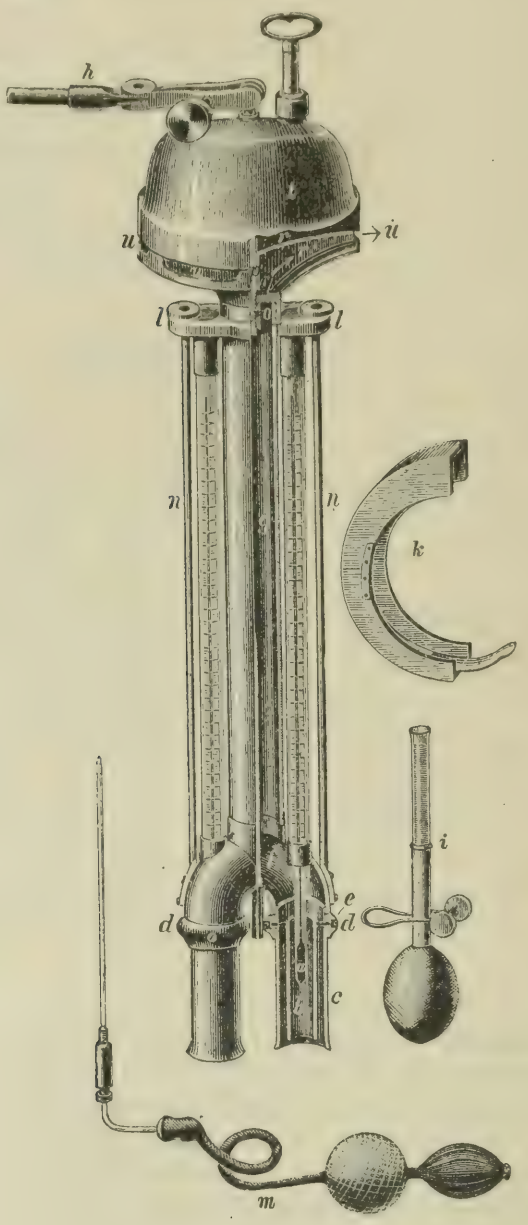


FIG. 11 ASPIRATION PSYCHROMETER

APPENDIX No. 2

PROOF OF FORMULA [5]

$$G = \frac{5284\ (t + 459.64)\ D_s}{P - e}$$

69 In this equation

G = grains of moisture per lb, of pure air at saturation
t = temperature of saturation in deg. fahr.
$t + 459.64$ = absolute temperature
D_s = density in lb. per cu. ft. of saturated water vapor at temperature t
= reciprocal of specific volume of steam
P = 29.92 = assumed standard of barometric pressure in in. of mercury
e = vapor pressure of saturated water vapor in in. of mercury
5284 = constant of the equation

70 According to the law of gaseous mixtures the total pressure is equal to the sum of the gaseous pressures of the component parts, and the weights of the two components are in proportion to the products of their respective pressures times their specific weights

$$\frac{W_1}{W_2} = \frac{p_1 S_1}{p_2 S_2} \text{ and } P = p_1 + p_2$$

For mixture of 1 lb. of air and water vapor saturated at a given temperature, therefore

$$\frac{W_s}{1} = \frac{S_w p_s}{1\ (P - p_s)}, \text{ or } W_s = \frac{S_w p_s}{P - p_s} \quad \frac{W_s}{1} = \frac{S_w p_s}{1\ (P - p_s)}, \text{ or } W = \frac{S_w p_s}{P - p_s} \quad \ldots [32]$$

where

Weight of air = 1 lb.
Specific weight of air = 1

$(P - p_s)$ = total barometric pressure in lb. per sq. ft.
W_s = weight of saturated water vapor in 1 lb. pure air
p_s = pressure of saturated water at given temperature t_1 in lb. per sq. ft.
S_w = specific weight of water vapor at temperature t, compared with air at same temperature and pressure; p_s or e
W = weight of moisture vapor in mixture containing 1lb pure air.

Also

$$S = \frac{V_a}{V_s} = V_a D_s \quad \text{.....[33]}$$

where

V_s = specific volume of saturated steam (volume of 1 lb.) at a given temperature, t

V_a = specific volume of air at the same pressure and temperature

D_s = density of saturated steam in lb. per cu. ft. at temperature t

But we have for air

$$\frac{PV}{T} = 53.35$$

$$V_a = 53.35 \left(\frac{t + 459.64}{P} \right) \quad \text{.....[34]}$$

Therefore

$$S = \frac{53.35\ (t + 459.64)}{p_s} D_s \quad \text{.....[35]}$$

Hence, substituting in [32]

$$W = \frac{53.35\ (t + 459.64)}{(P - p)} D_s \quad \text{.....[36]}$$

$$G = 7000\, W$$

$(P - e)$ in. mercury = (144×0.4908) $(P - p)$ lb. per sq. ft.

Hence

$$G = \frac{5284\ (t + 459.64)\ D_s}{(P - e)} \quad \text{.....[37]}$$

APPENDIX No. 3

DERIVATION OF FORMULA [6], GIVING THE EQUATION OF THE ADIABATIC SATURATION LINE

$$r'(W'-W) = C_{pa}(t-t') + C_{ps}W(t-t')$$

71 Assuming 1 lb. of pure air having the temperature t containing W lb. of moisture with the corresponding dewpoint t_1 and vapor pressure e_1 having a resultant adiabatic saturation temperature of t', assume also a moisture increment dW under adiabatic conditions resulting in a temperature increment of $-dt$. This moisture increment dW is evidently evaporated at a vapor pressure e_1 corresponding to temperature t_1 and superheated to temperature t. The temperature of the liquid is evidently constant at temperature t', from principle C. The total heat of the vapor in the increment is $H_1dW + C_{ps}(t\text{-}t_1)\,dW$, where H_1 is the total heat of steam corresponding to temperature t_1 and vapor pressure e_1, and $C_{ps}(t\text{-}t_1)dW$ is the heat required to superheat from saturation temperature t_1 to dry-bulb temperature t. The heat of the liquid evaporated, however, is $q'dW$ corresponding to temperature of saturation t'.

72 The total heat interchange required to evaporate dW under these conditions is therefore

$$\Sigma = [(H_1 - q') + C_{ps}(t-t_1)]\,dW \quad \ldots\ldots [38]$$

The change in sensible heat of 1 lb. of air and W lb. of water vapor due to the temperature increment $-dt$ is

$$\Sigma = -(C_{pa} + W C_{ps})\,dt \quad \ldots\ldots [39]$$

Since the change is adiabatic these values may be related by the equation

$$[H_1 - q' + C_{ps}(t-t_1)]\,dW - (C_{pa} + W C_{ps})\,dt = 0 \quad \ldots\ldots [40]$$

$$\int_W^{W'} \{H_1 - q' + [C_{ps}(t-t_1)]\}\,dW = \int_t^{t'} (C_{pa} + W C_{ps})\,dt \quad \ldots\ldots [41]$$

in which H_1 and t_1 are variables corresponding to the variable W while t is a variable related to W by the differential equation. A constant corresponding to t' is q' while C_{ps} may be taken approximately as a mean between its values at t_1 and at t' and C_{ps} as a mean between its values at t and at t'. The temperature of saturation is t'. and W' is the corresponding moisture content at saturation.

73 It is not necessary, however, to solve this equation in this form as this relationship may be simplified

$$\{H_1 - q' + [C_{ps}(t-t_1)]\}dW = \{H_1 - q' + [C_{ps}(t'-t_1)) + (C_{ps}(t-t')]\}dW \quad [42]$$

It may be shown thermodynamically, assuming steam to be a perfect gas, that

$$H_1 - q' + [C_{ps}(t' - t_1)] = H' - q' = r' \quad \ldots\ldots\ldots\ldots\ldots\ldots [43]$$

74 This may also be demonstrated approximately for the range of temperatures under discussion by computation from the values given in the steam tables of Marks and Davis, as in Table 8.

TABLE 8 COMPARISON OF ACTUAL VALUES OF r' WITH VALUES OF r' COMPUTED FROM THE TOTAL HEAT AT DIFFERENT TEMPERATURES t_1

t'	t_1	C_{ps}	H_1	q'	r' (Computed)
		$t' = 80$ *deg.*, $r' = 1046.7$			
80	70	0.44365	1090.3	48.03	1046.70
80	60	0.44356	1085.9	48.03	1046.74
80	50	0.44347	1081.4	48.03	1046.78
80	40	0.44338	1076.9	48.03	1046.6
		$t' = 100$ *deg.*, $r' = 1035.6$			
100	90	0.44401	1099.2	67.97	1035.67
100	80	0.44392	1094.8	67.97	1035.7
100	70	0.44383	1090 3	67.97	1035.64
100	60	0.44374	1085.9	67.97	1035.67
100	50	0.44365	1081.4	67.97	1035.81
100	40	0.44356	1076.9	67.97	1035.54
		$t' = 120$ *deg.*, $r' = 1024.4$			
120	110	0.44419	1108.0	87.91	1024.4
120	100	0.44410	1103.6	87.91	1024.57
120	90	0.44401	1099.2	87.91	1024.62
120	80	0.44392	1094.4	87.91	1024.64
120	70	0.44383	1090.3	87.91	1024.58
120	60	0.44374	1085.9	87.91	1024.50
120	50	0.44365	1081.4	87.91	1024.54
120	40	0.44356	1076.9	87.91	1024.47

$$r' = H_1 - q' + [C_{ps}(t' - t_1)]$$

or $$H' = H_1 + [C_{ps}(t' - t_1)]$$

75 Hence substituting in equation [41]

$$r' \int_W^{W'} dW + C_{ps} \int_W^{W'} (t - t')\, dW = C_{pa} \int_{t'}^{t} dt + C_{ps} \int_{t'}^{t} W dt \quad \ldots [44]$$

$$r'(W' - W) + C_{ps} \int_W^{W'} \left[\int_{t'}^{t} dt \right] dW = C_{pa}(t - t') + C_{ps} W \int_{t'}^{t} dt$$

$$+ C_{ps} \int_{t'}^{t} \left[\int_W^{W'} dW \right] dt \quad \ldots\ldots\ldots\ldots\ldots\ldots [45]$$

$$r'(W' - W) = C_{pa}(t - t') + C_{ps} W (t - t') \quad \ldots\ldots\ldots\ldots [46]$$

76 The same result may be obtained by equating the total heat in the air in any state with its total heat when in the state of adiabatic saturation. The total heat in a mixture of 1 lb. of pure air and saturated water vapor at a temperature t' calculated from a base temperature of 0 deg. fahr. and deducting the heat of the liquid, q', which as we have shown is unaffected by the adiabatic change, is

$$\Sigma = C_{pa}\, t' + r'\, W' \ldots\ldots [47]$$

77 The total heat under any other ad abatic condition, where temperature is t and moisture W, is

$$\Sigma = C_{pa}\, t + [\,(H_1 - q_1) + C_{ps}\,(t' - t_1)\,]\, W \ldots\ldots [48]$$

which is substantially equivalent to

$$\Sigma = C_{pa}\, t + r'\, W + C_{ps}\,(t - t')\, W \ldots\ldots [49]$$

Therefore since the change is adiabatic we may equate [45] and [47].

$$C_{pa}\, t + r'\, W + C_{ps}\,(t - t')\, W = C_{pa}\, t' + r'\, W' \ldots\ldots [50]$$

$$C_{pa}\,(t - t') + C_{ps}\,(t - t')\, W = r'\,(W' - W) \ldots\ldots [51]$$

where

$(t\text{-}t')$ = the true wet-bulb depression
$(W' - W)$ = the moisture absorbed per lb. of pure air when it is adiabatically saturated from an initial dry-bulb temperature t_0 and an initial moisture content W
C_{pa} = mean specific heat of air at constant pressure between temperature t and t'
C_{ps} = specific heat of steam at constant pressure between t and t'
r' = latent heat of evaporation at wet-bulb temperature t'

This is identical with equation [46] obtained by the differential method.

DISCUSSION

R. C. Carpenter said that he had carefully checked the results obtained by Mr. Carrier and had compared them with the old tables on this subject and with the newer data available, and he believed them to be a distinct advance on anything before presented, and also much more accurate.

L. S. Marks said that he had had some difficulty in reading the paper, because of some obscurity of phrasing by the author.

The fundamental thing about the paper appeared to him to be the establishment by direct experiment of the rational formula connecting the temperature indications of a psychrometer which is properly protected from external radiation. This rational formula, as the author states, was given in an imperfect form by Apjohn in

1836. It was developed even earlier than that, for August[1] understood the theory thoroughly in 1825, and deduced the best equation that was possible with the existing state of physical knowledge at the time. Regnault,[2] in his paper on Hygrometry, published in 1845, showed that the August equation did not give correct results, and that the wet bulb readings depended largely on the velocity of the air. He discredited the basic assumption of August's equation and of the equation now presented to us, because he thought that the air in contact with the wet bulb was neither reduced to the wet-bulb temperature, nor was fully saturated. It would appear from the experiments now presented to us that Regnault's conjectures on these points cannot be accepted, and that he must have underestimated the importance of the radiation factor. The rational theory presented here is complete only when applied to a psychrometer in which absorption of heat by radiation is prevented. Regnault's objection to it as applied to the psychrometer of his time, is justified by the facts, but not for the reasons which he advanced.

The basis of the rational theory is unfortunately obscured in the author's paper by his misuse of certain terms. He speaks of saturating air adiabatically. The action taking place at the wet bulb is in no sense adiabatic. It is exactly what takes place when bubbles of air at some temperature are sent through water at some lower temperature. It is an actual mixture of two substances at different temperatures. The thermal characteristic of this process, as carried out in a psychrometer without external radiation, is that the total heat of the two mixing substances is the same before and after mixing. By total heat is meant the quantity:

$$H = U + Apv$$

where

U = the internal energy

$A = \frac{1}{778}$

p and v = specific pressures and volumes respectively

The author's principal equation is simply the writing down of this relation between the total heats.

The rest of the paper seems to be devoted principally to deducing special equations for saturated and superheated steam at low pressures, on the assumption that they behave as perfect gases. Table 6 shows that that assumption is satisfactory.

[1] Pogg. Ann., 5, 69. 1825.

[2] Ann. de Chim. et de Phys., 3, xv, 201. 1845.

The suggestion that the psychrometer might be used as an instrument for the determination of the specific heat of air or other gases was also made by Regnault in 1845.

The results obtained by the author form a solid contribution to the science of hygrometry, and give more confidence for the future in the use of such an instrument as the aspiration psychrometer. Professor Marks had had occasion to check up the author's hygrometric charts, and had found them quite accurate.

O. P. Hood objected to the statement, that "the direct use of steam" for humidifying is of " little engineering interest or value." He thought it true that the greatest quantity of air in the country needing to be conditioned would be that connected with mining interests, particularly where very large quantities of air are handled. In winter the most feasible plan so far found is this introduction of steam direct into the current of air. While the method is extremely simple, almost crude, and there are many objections to its use in mines, still the proper humidifying of air, so as not to add too much and to add it at the proper time, is a question interesting to mining engineers.

G. A. Goodenough (written).[1] The chief criticism of the paper relates to the manner in which the material is presented. The paper is difficult reading, even to one fairly familiar with thermodynamic principles. Such terms as temperature of adiabatic saturation, per cent of isothermal saturation, etc. are not familiar to most of us. There is a lack of precision in the use of terms, such as specific weight, and density. The symbols employed are not conducive to clearness.

The validity of the final results rests upon the truth of the principles stated in Par. 30 and upon the establishment of the relation expressed by equation [6]. There need be no question regarding principles (A) and (B). To make the statements somewhat clearer reference may be made to Fig. 12. The point A represents on the temperature-entropy plane the initial state of the moisture in the air at the absolute temperature T. The line FEA is a line of constant pressure e_1, and T' is the absolute temperature corresponding to the dewpoint. As more vapor is introduced the vapor pressure rises, but the temperature falls; hence the point that represents the state of the vapor lies below the line T but above the curve EA. As

[1] Prof. of Thermodynamics, Univ. of Ill., Urbana, Ill.

the process is continued the state point moves downward and to the left along some such curve as AB and finally reaches the saturation curve at point B. The air is now saturated and the temperature T' at B is the so-called temperature of adiabatic saturation. According to principle (C), this temperature T' is precisely the temperature assumed by the water that is evaporating, i.e., the wet-bulb temperature. This interesting principle seems to be amply verified by the experiments described in Appendix No. 2.

The demonstration given in Appendix No. 4 may now be examined and certain obscurities may be removed. Let the weight of water

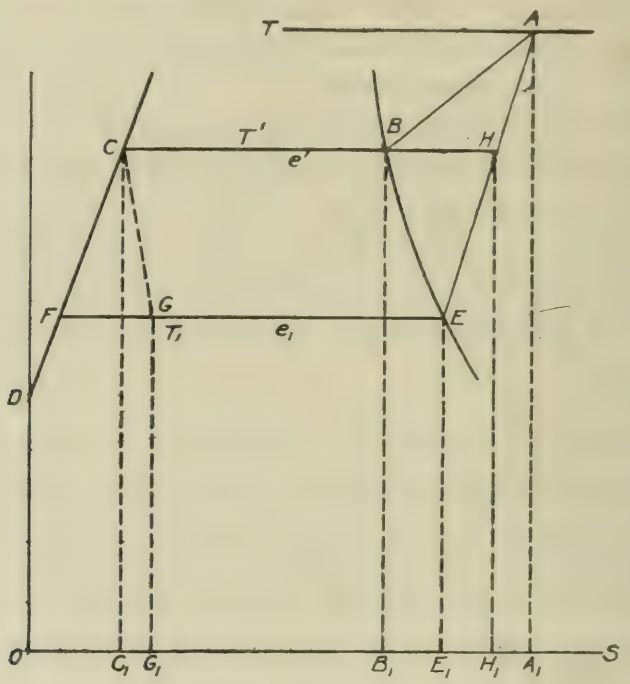


FIG. 12 PRESSURE TEMPERATURE DIAGRAM

dW originally in the state denoted by point C be introduced into the mixture of air and vapor. The vapor pressure falls from the value e' to e_1, the process being represented by the curve of constant heat content CG. Further vaporization of the liquid is represented by GE and superheating by EA. The heat absorbed during the process is represented by the area G_1GEAA_1 = area $ODFEAA_1$ − area $ODFGG$. But area $ODFEAA_1$ represents the total heat corresponding to state A, and area $ODFGG_1$ = area $ODCC_1$ represents the heat of the liquid q' corresponding to state C. Hence the heat absorbed is

$$[H_1 + C_{ps}\,(T - T_1) - q']\,dW$$

as given in equation [40]. If the whole process is adiabatic, this heat must be received from the air and steam already present and the resulting equation is

$$[H_1 - q' + C_{ps}(T - T_1)]\,dw + (C_{pa} + WC_{ps})\,dT = 0$$

This is the same as equation [42] except that the sign between the two terms is positive as it clearly should be, since dT is intrinsically negative. The simplification of this equation by means of the relation expressed by equation [45] is exceedingly ingenious, and worthy of some attention. Referring to Fig. 12, equation [45] asserts that area C_1CBB_1 $(=r^1)$ = area G_1GEHH_1. Since the heat content at G is by hypothesis equal to the heat content at C, this is equivalent to the assumption that the heat content at H is the same as that at B. For a perfect gas, B and H lying on the same isothermal, this assumption would be strictly true; and for superheated steam, provided B and H are sufficiently close, the relation is sufficiently well satisfied. Combining the relation [45] with the preceding equation the result is:

$$r'dW + C_{ps}(T - T')\,dW + C_{pa}\,dT + C_{ps}\,WdT = 0$$

whence

$$r'\int dW - C_{ps}T'\int dW + C_{pa}\int dT + C_{ps}\int (TdW + WdT) = 0$$

This equation takes the place of the author's equation [46]. On account of the error in sign in equation [42] an inexact differential is contained in [46] and integration is therefore impossible without a relation between the variables T and W. The passage from [46] to [48] involves some manipulation that would not be sanctioned by mathematicians. With the proper signs, however, the differential is exact as shown in the third equation and the integration is readily effected with the following result:

$$r'(W' - W) - C_{ps}T'(W' - W) + C_{pa}(T' - T) + C_{ps}(T'W' - TW) = 0$$

This equation is readily reduced to the desired form

$$r'(W' - W) = C_{pa}(T - T') + C_{ps}W(T - T')$$

Having the fundamental equation deduced by sufficiently rigorous processes, the remaining relations, equations [7] to [28], follow without difficulty. The use of Swann's expression for the specific heat of air is to be commended. The expression for the latent heat of steam is also satisfactory, though possibly the resulting values

will ultimately be found slightly low. The expression for the specific heat of steam under the conditions assumed, namely

$$C_{ps} = 0.4423 + 0.00018t$$

is certainly somewhat in error. For the low pressures under consideration, the expression for specific heat may be taken as

$$C_{ps} = 0.42 + 0.0001t$$

However, as C_{ps} drops out of consideration between equations [19] and [20], the final result is in no way affected by a slight change in C_{ps}.

Equation [23] may therefore be accepted as substantially accurate, and the psychrometric charts, Fig. 1 and 2, may likewise be accepted as most valuable aids in the solution of psychrometric problems.

R. C. H. Heck (written). The matter presented in this paper is decidedly valuable and interesting and the term adiabatic saturation is especially well chosen. But the underlying theory of the subject, as developed in Appendices Nos. 3 and 4, can be much shortened and simplified, to the greater ease of the reader in following it. For the confused and confusing deduction in Appendix No. 3, proof of formula [5], may be substituted the following:

With a total pressure of P lb. per sq. in., of which $(P - p)$ is exerted by air and p by steam or vapor, we consider 1 lb. of air and W lb. of steam. At ordinary temperature t or absolute temperature $T = 459.6 + t$, the volume in cu. ft. of the pound of pure or dry air is

$$V_a = 53.35 \frac{T}{P - p}$$

If p and t correspond as pressure and temperature of saturated steam, 1 lb. of steam will have a volume of V_s cu. ft., or a density $D_s = \frac{1}{V_s}$ lb. per cu. ft., these quantities coming directly from the steam table. The weight W will then fill a volume of WV_s or $\frac{W}{D_s}$ cu. ft., which volume must be the same as V_a, since air and vapor are diffused through the same space, Therefore

$$\frac{W}{D_s} = \frac{53.35\,T}{P - p}$$

and this leads at once to formula [5], the change in units of measurement being a mere detail.

The derivation of formula [6] can also be much abbreviated and clarified. Restating conditions, 1 lb. of dry air with an original content of W lb. of vapor (less than saturation) passes over a water surface and takes up all the moisture that it can evaporate by virtue of the heat available. Assuming that no heat comes from any source except the air current and the limited mass of water upon which it acts, and that the action has persisted long enough for conditions to become stable, the temperature of adiabatic saturation will be established. That the water settles to this temperature even when there is a large excess of air is an experimental fact rather than a conclusion which could have been arrived at by reasoning. With such excess of air, as in the case of the psychrometer, the small amount that is or would be saturated is immediately mixed with a much larger amount of the original air, so that the resulting average vapor content and temperature of the whole current are far from W' and t' being very little changed from W and t. If it is true that the water is influenced wholly by the thin layer of air directly in contact with it, this air may be thought of as isolated from the rest of the current. Passing over the intimate detail of the process of heat transfer, there is at once an equation between heat absorbed by the water in vaporization at t' and heat given up by the air in cooling original t to t', so that

$$r' (W' - W) = (C_{pa} + WC_{ps}) (t - t')$$

A question arises here which concerns not the form of expression of Mr. Carrier's theory but its fundamental assumption. May not the large excess of relatively warm air flowing past the wet bulb of the psychrometer act as a "surrounding body," supplying a little heat to the water by radiation? Undoubtedly, rapidity of the vaporization induced by exposure to air depends far more upon the dryness of the air than upon purely thermal conditions; in other words, the difference between original vapor tension and that of saturation has far greater influence than a temperature difference such as hastens conduction of heat. Further, the radiating power of a body of gas is very weak. For these reasons, the heat received from the warm air which takes up no new moisture will be small in comparison with that received from the immediately adjacent layer of air which becomes saturated. But may not this extra heat account for the small increase, above the long accepted value, in the specific heat of air as determined by the experimental method described in the paper?

There is positive error as to the idea of physical operation, that water of temperature t' is evaporated at some other and lower temperature t_1, and the vapor then superheated past t' to t. No matter what the pressure and composition of the superincumbent atmosphere, vapor can be evolved only at the temperature of the liquid from which it is rising, and with the saturation pressure or tension belonging to that temperature (Fig. 13).

It is entirely correct, however, to say that there is essential equivalence as to total heat between evaporation at t' and evaporation at a lower temperature t_1 plus superheating to t'. With steam pressure as ordinate and total heat as abscissa, the slightly inclined lines in Fig. 13 are isothermals or lines of uniform temperature; the curve through their upper ends, along which the degree numbers are placed, is the saturation line. These isothermals show how the total heat up to a certain temperature varies with the pressure (a constant in any particular operation) at which this heat is imparted. For a perfect gas, the isothermals of total heat would be verticals, indicating constancy of h; their inclination in this diagram measures the departure of steam from ideal gaseous properties (Fig. 14).

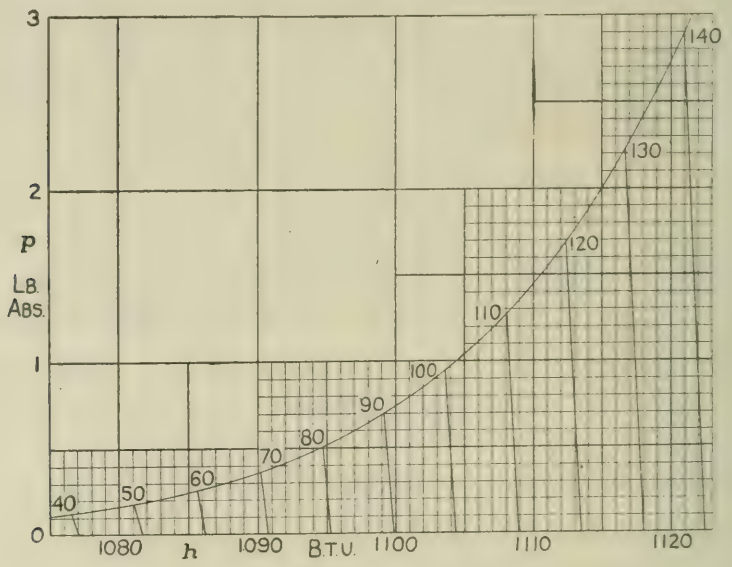


FIG. 13 ISOTHERMAL CHART

A graphical illustration of the integration represented by equations [46] and 47] is very helpful. It is better to distinguish initial conditions by a subscript, calling them W_o and t_o. Then, if instead of complete cooling from t_o to t', with vapor increase from W_o to W', action to some intermediate limits W and t be considered, equation [46] may be written

$$r'(W-W_o)+C_{ps}\int_{W_o}^{W}(t-t')\,dW=C_{pa}(t_o-t)+C_{ps}W_o\int_{t}^{t_o}dt+$$

$$C_{ps}\int_{t}^{t_o}(W-W_o)\,dt$$

The heat quantities belonging to the superheated vapor of varying

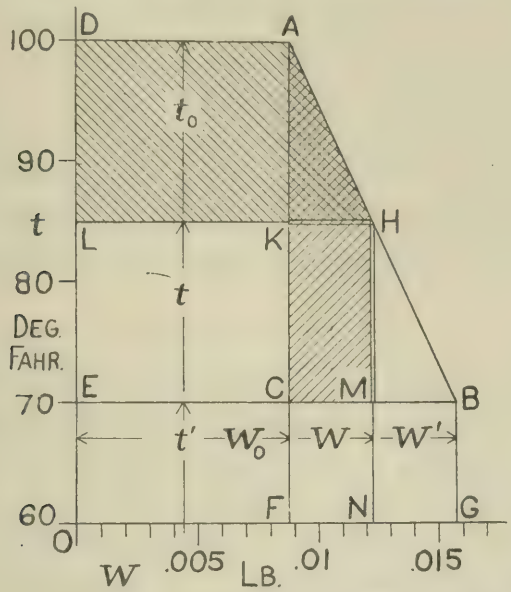


FIG. 14 HEAT OF SUPERHEATED STEAM DIAGRAM

weight W are illustrated in Fig. 14. The following mathematical statements seem hardly to call for verbal supplementation:

$$(t-t')\,dW = \text{strip like } HM$$

$$\int_{W_o}^{W}(t-t')\,dW = \text{area } AHMC$$

$$W_o\int_{t}^{t_o}dt = \text{area } DAKL$$

$$(W-W_o)\,dt = \text{strip like } KH$$

$$\int_{t}^{t_o}(W-W_o)\,dt = \text{area } AHK$$

The partial cooling and saturation just described might represent the average condition, at some point in transit, of the current of air flowing through a drying kiln. Here again the minute detail represented by the differential expression and its integration may be passed over, and it is a simple matter to write directly the equation of net effect

$$r'(W - W_o) + C_{ps}(t - t')(W - W_o) = C_{pa}(t_o - t) + C_{ps} W_o (t_o - t)$$
(area $CKHM$) (area $DAKL$)

The left-hand member of this equation shows evaporation of $(W - W_o)$ plus superheating t_o to t, the right-hand member shows cooling of the original "air" mixture from t_o to t.

Of the general operation of inducing vaporization by air currents, adiabatic saturation is a particular, limiting case, in that all heat is excluded except what is carried by the original air. The diametrically opposite condition is exemplified in the working of the cooling tower, where heat for vaporization is all taken from the descending current of warm water. The ascending air-and-vapor current actually rises in temperature, so that the final saturation capacity is that of air at, or nearly at, the initial temperature of the warm water coming to the tower.

Thos. M. Gunn (written). In this paper is to be found without doubt a valuable asset to the engineer concerned with computations on atmospheric humidity and corresponding heat changes. It is to be noted that the data and charts have particular application to a barometric pressure of 29.92 in. of mercury, and are, therefore, not directly applicable at high altitudes, or to problems where the pressure of the air is varied from standard atmospheric in the course of its treatment. There is still need of broader treatment of the subject for these varying conditions. Moreover, recent developments in internal-combustion engines and gas turbines show a great need of the investigation of laws pertaining to the adiabatic expansion of vapor-gas mixtures, especially within the range where condensation occurs.

The question of definition should be carefully considered. The expressions, per cent of isothermal saturation, per cent of adiabatic saturation, and absolute humidity are fully defined in this paper. At the same time other terms are used which are more or less synonymous with one or another of these, but not being definitely stated lead to doubt or ambiguity. Such terms are per cent or degree of saturation, humidity, and relative humidity. For instance, the term percentage humidity as it appears on Figs. 1 and 2 evidently refers to per cent of isothermal saturation, while it is not clear that the expression has the same meaning throughout the paper. The difficulty, however, is not as great in this paper as in most literature on the subject.

The curve of total heat is stated in Par. 70 to represent the sum of sensible heat of the air above 0 deg. fahr. and the latent heat of the

vapor, but not the heat of the liquid. In calculations upon the exchange of heat between air and water in a cooling tower, or in the case cited of removal of moisture by refrigeration, the heat of the liquid should not be neglected if accuracy is required. In other words, such a process as takes place in the cooling tower does not correspond to what Mr. Carrier defines as adiabatic saturation. To have the heat given up by the cooling of air and water vapor equal that used in the process of evaporation, or in other words to fulfil equation [6], there must be no communication of heat to or from the acting elements. If the water is in contact with other water of another temperature there is such a transfer. Where the original temperature of water exposed to such evaporation is other than the so-called wet-bulb temperature, the heat of the liquid must be taken into consideration.

This would make it appear that if the water with which a wet bulb is moistened is other than the correct wet-bulb temperature, the heat of the liquid and of the thermometer bulb itself will prevent the bulb temperature reaching its correct value. The process at first is not adiabatic when the psychrometer is first whirled, but is such as to bring the temperature rapidly toward the point of adiabatic saturation as a limit, and it is only a question as to whether the moisture on the bulb will not all be evaporated before the required accuracy has been reached, which does not often occur.

It appears that equation [6] might be questioned, even assuming the definition of adiabatic saturation to be permissible. In it the latent heat of that amount of water which is evaporated is to be equated to the change in heat contents of the original air-vapor mixture. The latter is said to be equal to

$$C_{pa}(t-t') + C_{ps}(t-t')$$

which would be correct, if the pressure of the air and of the water vapor remained constant. This is not the case. It is assumed that the total, or barometric, pressure remains constant. The formation of more vapor causes an increase in vapor pressure, and this in turn requires that the partial pressure of the air decrease. Then the partial pressures of the air and of the original vapor are both varying, and the heat change will not be given with even approximate correctness by the above expression which assumes constant pressure, *unless* the pressure change is small. It is small within the lower ranges of temperatures such as those represented in Fig. 1, but at higher temperatures, the act of saturating air with moisture incurs

large changes in the partial pressure of the air, and, therefore, I question the correctness of this formula for this range, and the correctness of the adiabatic saturation lines at the higher temperatures indicated in Fig. 2.

THE AUTHOR. I agree fully with most of the points brought out in the discussion. There are a few points, however, which seem questionable.

Professor Heck suggests: "May not the large excess of relatively warm air flowing past the wet bulb of the psychrometer act as a 'surrounding body' supplying a little heat to the water by radiation?" Apparently he contends that a form of gaseous radiation exists which would cause the wet-bulb temperature to be higher than the true temperature of adiabatic saturation. In answer to this, I would refer particularly to the experimental evidence given in Appendix No. 1, Table 6, which is from a test in which the protected wet bulb subjected to a current of dry air was compared with the readings of a wet sling psychrometer bulb under ordinary exposure. In Table 7 we have a log of a similar test except that instead of having a protected bulb subjected to a current of dry air, we gave the same bulb subjected to a current of air adiabatically saturated before coming in contact with the bulb. If anything, the temperature of the adiabatically saturated air seems to be slightly higher than the corresponding temperature of the protected wet bulb, but this difference is so slight as to be negligible. The large number of tests made with practically the same results would seem to thoroughly substantiate the truth of (*C*), Par. 22, which is in question. Furthermore, from the theoretical side, Professor Heck admits that the radiation from a gas must be relatively very slight and, by referring to Table 6, Appendix No. 1, and to Fig. 6, it will be seen that the radiation error from external objects lay between 1 and 2 per cent, depending upon the velocity.

However, it would seen that the radiation of the gas, if any, of which Professor Heck speaks, must be very much smaller than this. Possibly not more than 1 per cent of this, that is, the error due to radiation of the gas might possibly be about one-fiftieth of 1 per cent of the total depression and entirely beyond the range of ordinary instruments. Professor Heck suggests further: "May not this extra heat account for the small increase, above the long accepted value, in the specific heat of air as determined by the experimental method described in the paper?" The value for the specific heat of air given in the paper was not determined by the experimental method herein described,

but by an entirely independent method of apparently great accuracy, by Swann (see Par. 30). The formerly accepted value of the specific heat of air is certainly a great error.

The author believes Swann's value of specific heat of air is considered more nearly accurate by physicists than the old value. The author, however, is using the method suggested in his paper for checking this value of specific heat as applied to in the psychrometric tables.

Professor Heck objects to the statement in Par. 71, Appendix No. 3, that the water is evaporated at a vapor pressure corresponding to the temperature t_1, that is, the dewpoint, his claim being that evaporation can take place only at the temperature and pressure corresponding to the temperature of the liquid. This contention is undoubtedly true, but the author's statement is also correct.

It is evident that the vapor pressure is at once reduced from the pressure corresponding to the wet-bulb temperature to the pressure corresponding to the dewpoint at a lower temperature. In fact, it is the difference in vapor pressure between the dewpoint and the wet-bulb temperature that causes the evaporation. In any event, the ultimate result is to evaporate a particle of water at an ultimate pressure corresponding to the dewpoint and with a superheat corresponding to the difference between the dry-bulb temperature and the dewpoint temperature, exactly as the author has stated.

Mr. Gunn emphasizes the necessity of taking into account the heat of the liquid when dehumidifying air by means of a cold water spray, or in heating. This has not been taken into account in the total heat curve for a very good reason. It always has to be calculated independently, because invariably the condensed moisture will be carried through a different range form the air. The heat of the liquid, however, is but a small percentage of the total heat so that only a small percentage correction is necessary.

In another paragraph Mr. Gunn states: "This would make it appear that if the water with which a wet bulb is moistened is other than the correct wet-bulb temperature, the heat of the liquid and of the thermometer bulb itself would prevent the bulb temperature reaching its correct value," etc.

Now as a matter of fact, it is quite a difficult matter to bring the temperature of the wet bulb down to the proper amount on account of the heat of the liquid without having it become entirely dry, and the only sure way of obtaining a correct wet-bulb reading is to use water which is very close to the proper wet-bulb temperature.

Mr. Gunn also questions the correct physical basis of equation [6].

on the ground that it does not take into account the decrease in the partial pressure of the air and the increase of the partial pressure of the water vapor, as the water is evaporated. He states it will introduce an error which will be more noticeable at the higher temperatures and humidity. In this argument Mr. Gunn is quite mistaken in his premises. The change in partial pressure of the air and of the water vapor has absolutely no effect upon the amount of heat required, so long as the total barometric pressure remains constant. Mr. Gunn apparently overlooks the fact that the increase in volume is directly in proportion to the decrease in partial pressure of the air, and that so long as the product of volume and partial pressure is constant, the change in partial pressure will have no effect on the heat in the air. The effect is purely that of dilution and is exactly the same as though a different gas were introduced at constant barometric pressure. In this case the partial pressure of the air would be reduced owing to a mixture with the other gases, but if the total barometric pressure remains unchanged, it is evident that no heat change will take place.

There is one feature in the paper which seems open to criticism, and which has not been mentioned by any one. I believe that much further investigation is necessary before the results can be fully accepted. In fact, until this investigation is made it is to my mind impossible to have complete assurance of the accuracy of the tables of weights of water vapor contained in 1 lb. of saturated air. This feature to which I refer is the specific weight of water vapor in a mixture of air and water vapor. In Fig. 4 is given the curve of the specific weight of water vapor as calculated from the tables of Marks and Davis. This deduced from experimental data in which the experiments were conducted upon a pure vapor of water, and it has been assumed that they apply directly to the mixture of water vapor and air at corresponding temperatures. This assumption does not seem to me wholly warranted without experimental corroboration. Referring to the curve in Fig. 4, we see that the specific weight of steam increases with and is a function of the vapor pressure. The cause of this increase in specific weight seems to be due to an increased compressive action at the increased pressure of the water vapor which gives a greater relative density at the higher pressures than would correspond to that of a true gas. Now the question arises whether in a mixture of air and water vapor, the specific weight is effected by the pressure of the water vapor alone, in which case the specific weight would be unaffected in the mixture, or whether the

partial pressure of the air has an effect corresponding to an increased vapor pressure.

For example, pure water vapor at 80 deg. has a specific weight of 0.6240. Corresponding to atmospheric pressure and a temperature of 212 deg., it has specific weight of 0.6330. Now on the assumption that its weight is affected only by the vapor pressure, we would have in a mixture of air saturated with water vapor at 80 deg. a specific weight of 0.6240; but on the assumption that it was affected by the total pressure and not by the vapor pressure, we would have at 80 deg. a specific weight exactly the same as that at 212 deg., or 0.6330. This we see would permit a pound of air saturated at 80 deg. to hold $1\frac{1}{2}$ per cent more than according to our present theory.

Whether or not the partial pressure of air in a mixture of air and water vapor has any effect upon the specific weight of the water vapor, can be determined experimentally by introducing into a vessel under a vacuum water vapor containing a slight degree of superheat and determining its true dewpoint by means of the dewpoint instrument. Then air which is substantially free from water vapor may be added, by evaporating and superheating liquid air for example, thus raising the total barometric pressure without increasing the vapor pressure. After a certain amount of pure air has been added, the dewpoint may again be determined, maintaining practically the same degree of superheat as before, and if it is true that the specific weight of water vapor is determined only by the vapor pressure, then there will be no change whatever in the dewpoint observed. On the other hand, if it is affected by the partial pressure of the air, then the dewpoint of the air will be correspondingly reduced.

No. 1341

AIR-CONDITIONING APPARATUS

PRINCIPLES GOVERNING ITS APPLICATION AND OPERATION

By Willis H. Carrier, Buffalo, N. Y.
Associate Member of the Society
and
Frank L. Busey,[1] Buffalo, N. Y.
Non-Member

Air conditioning is a term which may be generally applied to the positive production and control of desired atmospheric conditions within an enclosure, with respect to moisture, temperature and purity. Its particular concern is the regulation of humidity.

2 The fundamental principles underlying the art of air conditioning have already been discussed in an accompanying paper.[2] It is the purpose of the present paper to describe the apparatus employed and to submit comprehensive data relating to its practical application and operation. While much of this material pertains to humidifying and humidity control, some important original data are given on heat transmission by convection, which is an important factor in many air-conditioning processes. Among these may be mentioned the heating of air by passing over steam coils; the simultaneous heating and cooling of air, as in a heat interchanger as employed in dehumidifying; and the dehumidifying and cooling of air with surface condensers. Upon these data as a basis a new theory of heat transmission by convection is established. A rational formula is derived embracing both the effect of velocity and temperature difference with a correction for variation of air density in the surface film.

[1] Engineering Experiment Sta., Univ. of Ill.

[2] Rational Psychrometric Formulae, Trans., vol. 33, No. 1340.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

3 Air-conditioning apparatus for controlling the humidity of air may be broadly classified, according to use, into humidifiers proper, which add moisture to the air in required amounts; and dehumidifiers, which remove a variable quantity of moisture from the air to reduce it to the required standard. The relative humidity of air

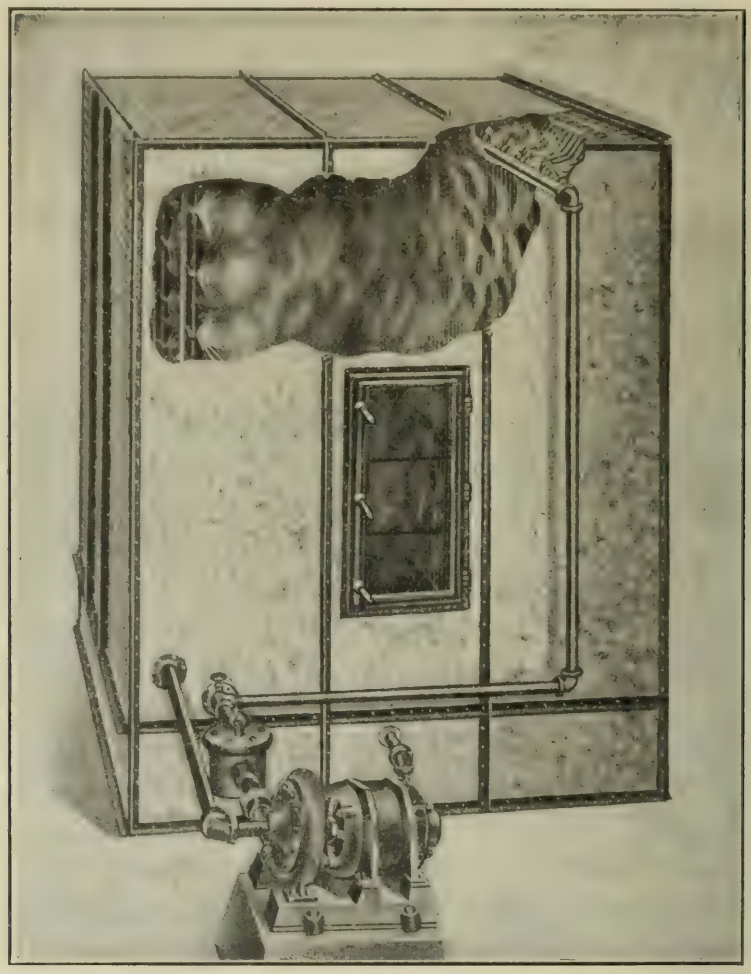


FIG. 1 AIR WASHER

may also be altered, and in a measure regulated, simply by changing its temperature without affecting its moisture contents.

TYPES OF HUMIDIFIERS

4 Humidifiers may be classified into the spray and evaporative types, the latter being divided again into direct and indirect. The humidity of the air may also be increased by the direct intro-

duction of steam into the air supply or into the room. Since the total heat of the vapor at atmospheric temperature is somewhat less than the total heat at steam temperature, this raises the temperature of the air perceptibly and is therefore intolerable in the majority of cases. Added objections to the direct use of steam are that it frequently gives a noticeable odor and that it is difficult to

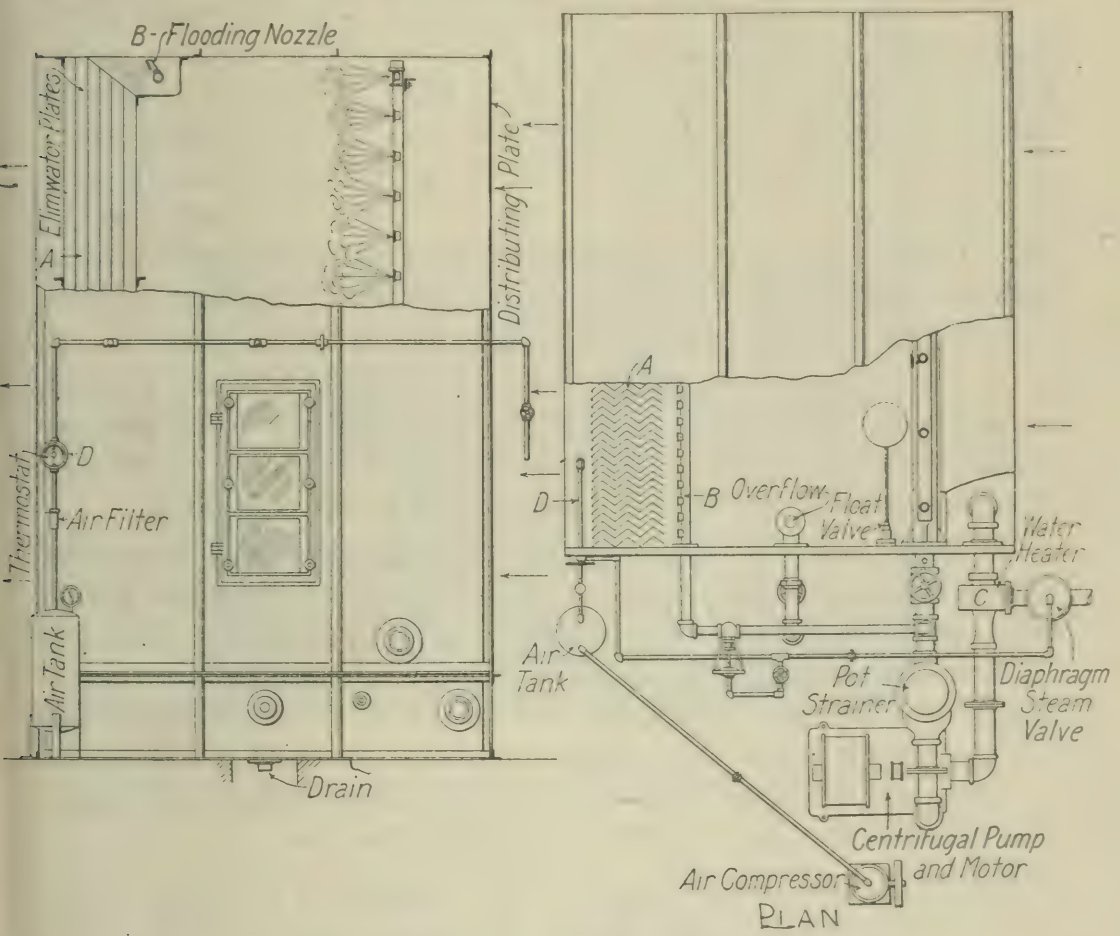


FIG. 2 AIR WASHER WITH HUMIDITY CONTROL

regulate. Its use is of so little engineering interest or value that it need not here be considered.

5 The spray and evaporative types of humidifiers have a distinct value aside from humidifying in their possession of a cooling effect which is in direct proportion to their moistening effect. The direct spray type of humidifier is distinguished from the evaporative type in that it introduces a finely divided or atomized spray directly into the room in constant volume, while the evaporative type introduces only the water vapor. There is also a mixed type which discharges both moist air and free moisture into the room.

6 In what may be termed the indirect evaporative humidifier the air is partly or entirely taken from the outside and is humidified and conditioned before it is introduced into the room. In the direct evaporative type the water vapor passes directly into the air of the room. The indirect system of air conditioning is also termed the central system.

7 A disadvantage of the direct spray type is that it always introduces a fixed quantity of moisture regardless of the needs or condition of the room until it is closed off by hand, or through a separate automatic control. In the evaporative type, on the contrary, there is an inherent self-regulating feature owing to the fact that the rate of evaporation is in direct proportion to the moisture deficit in the air. This is especially true in the indirect evaporative type, which, with all outside air, will maintain an absolutely uniform relative humidity, other conditions remaining constant.

8 Since the indirect evaporative type is capable of the greatest development and widest application, and therefore is of greatest engineering interest, this paper will be devoted largely to the theory of operation of this system and methods of its application.

9 Figs. 1 and 2 show the type of air purifier and humidifier adapted for use in the ventilation of auditoriums, offices and public buildings. Its primary object is the removal of impurities from the air, but combined with this is the very important function of regulation of the humidity, when below the minimum standard. When the humidity is above the required minimum standard it is designed to wash the air without objectional increase in its moisture contents. The essential features of this type of apparatus are:

a A distributing plate for the purpose of reducing eddies and distributing the air uniformly over the area of the washer.

b A system of atomizing sprays so arranged as to fill the air completely with water particles uniformly distributed over the chamber area.

c A centrifugal pump for maintaining the proper pressure on the spray nozzles.

d A settling chamber provided with proper strainers for the removal of dirt from the spray water.

e An eliminator, *A*, of proper construction and ample surface for washing the air by impact and centrifugal force and for the removal of all free moisture (Fig. 3).

f Flooding nozzles, *B*, to distribute an additional amount of water on the eliminators to increase the amount of available wetted surface, to flush the eliminators and to provide a means of washing the air without greatly affecting the humidity.

g Automatic water heater, *C*, for supplying heat and moisture to the air through the water spray. This may be either of the closed type or of the open, ejector type.

h Dewpoint thermostat at *D*, subject to the temperature of saturation and connected to motor valves controlling the supply of heat to the spray water.

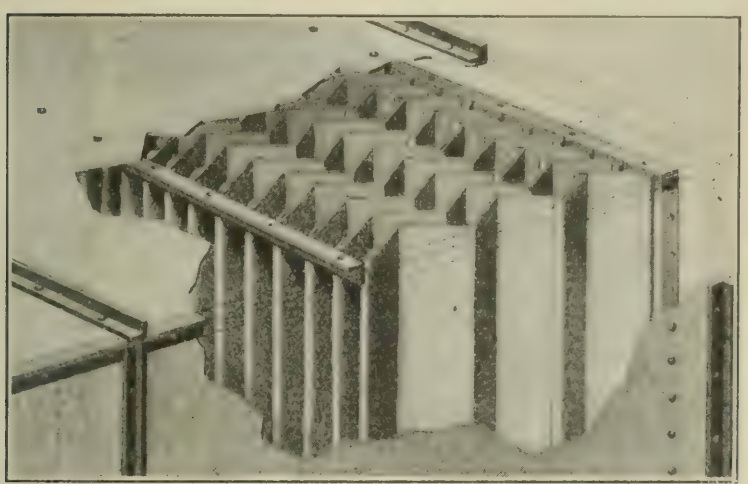


FIG. 3 DETAIL OF ELIMINATOR FOR REMOVING FREE MOISTURE

10 Air is drawn through this humidifier by means of a centrifugal fan at a velocity of about 500 ft. per minute. The temperature of the air is raised immediately in the humidifier from any outdoor temperature to that necessary to hold the desired amount of moisture, ordinarily to about 40 deg. fahr.

11 The indirect humidifier designed for industrial application is shown in Figs. 4 and 5. This is intended primarily to deliver completely saturated air, even when the saturation temperature is above the minimum point. This condition of saturation and the cooling effect are of primary importance; the cleaning effect being usually of secondary consideration. The industrial humidifiers differ from the public building air washer in the following particulars (Figs. 6 and 7):

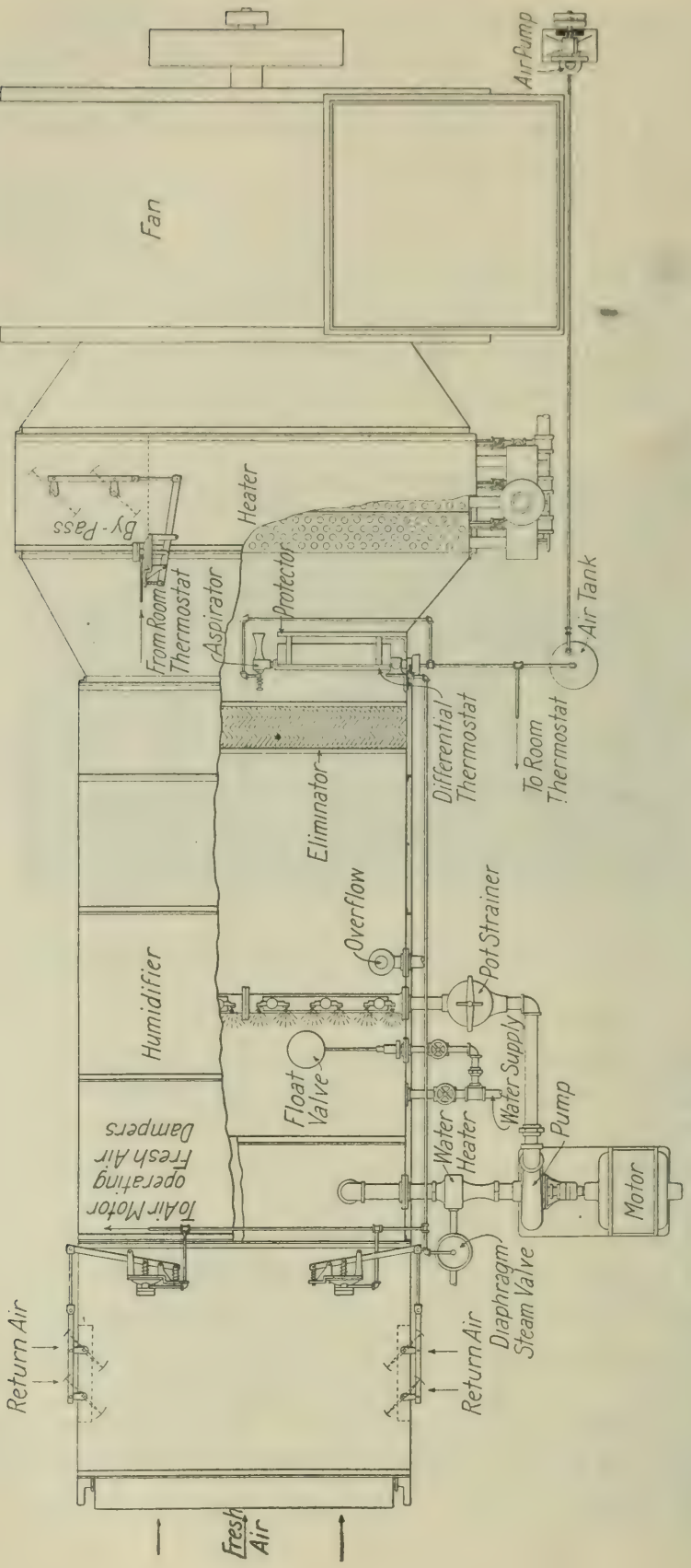


FIG. 4 AIR WASHER WITH HUMIDITY CONTROL—PLAN

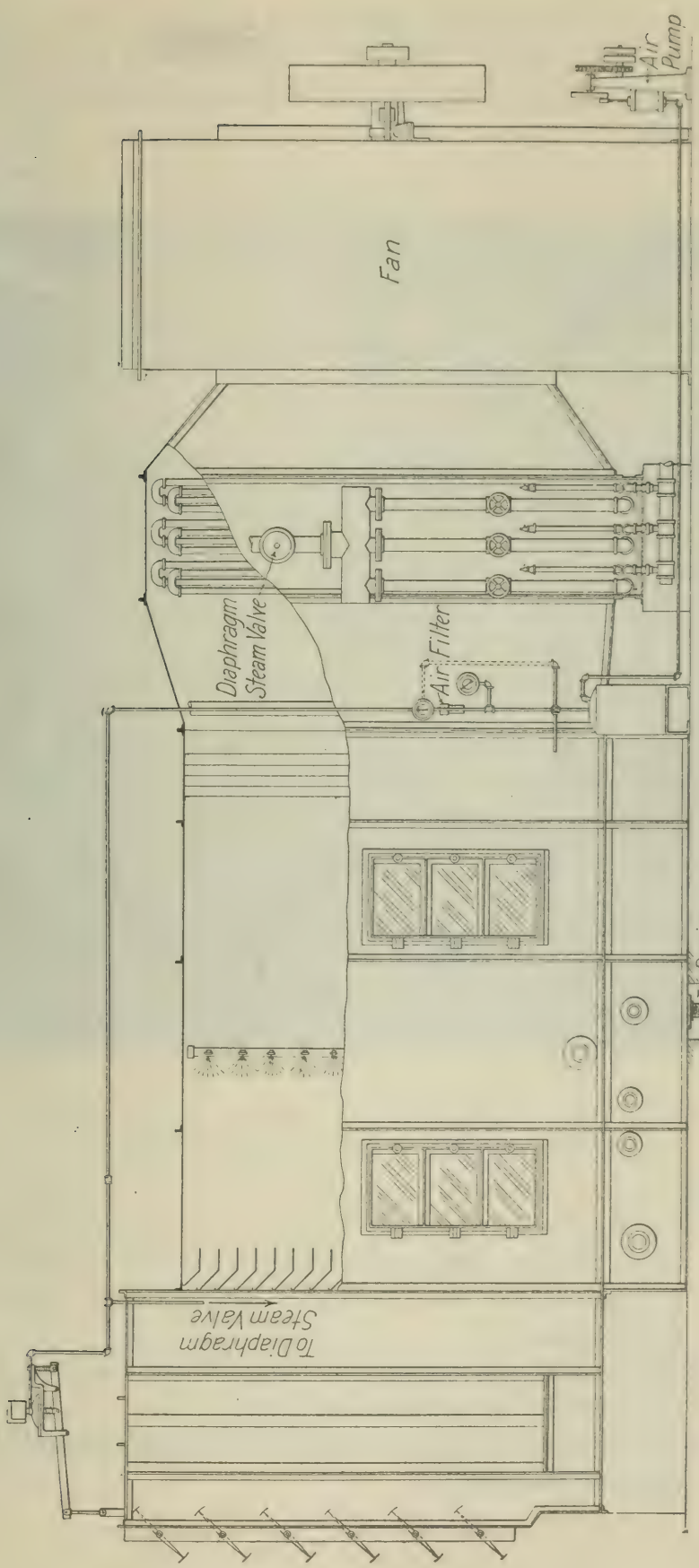


FIG. 5 AIR WASHER WITH HUMIDITY CONTROL—ELEVATION

a The centrifugal atomizing sprays are more numerous, and operate at higher pressure to give a more finely divided mist, and discharge in the opposite direction to the air flow.

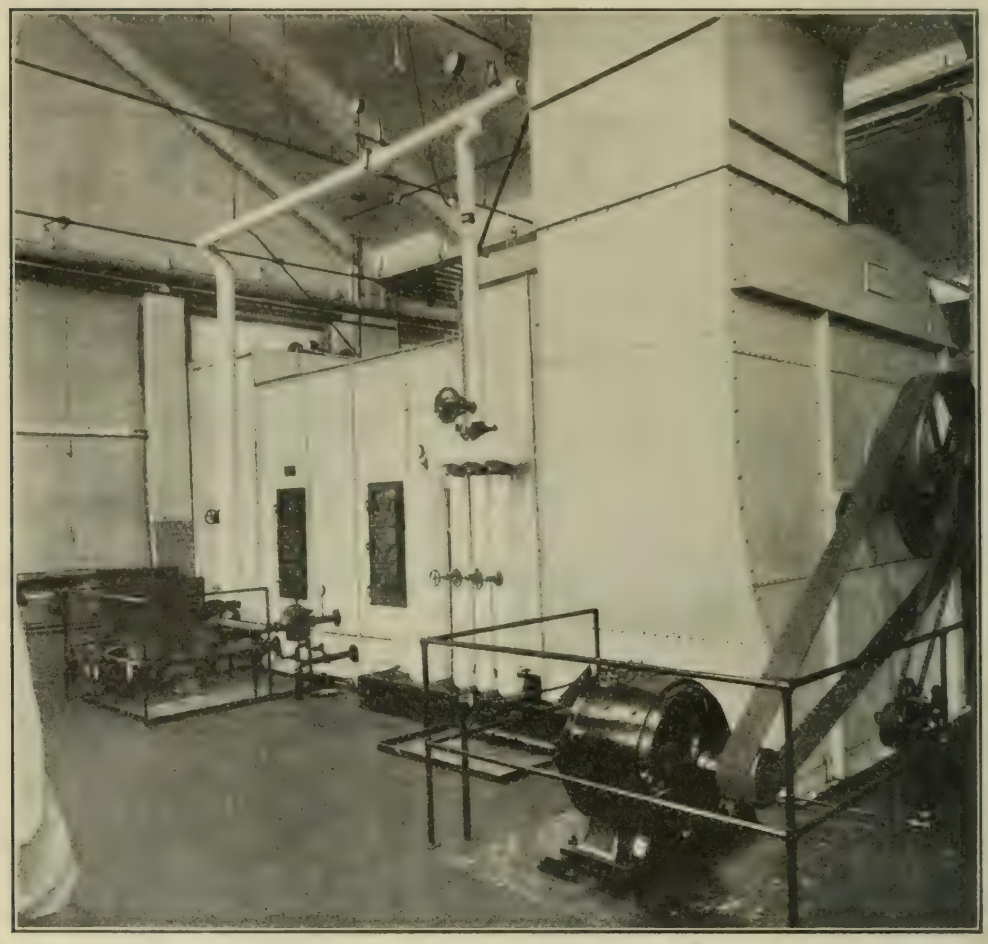


FIG. 6 GENERAL VIEW OF HUMIDIFIER

b The distributing plate is replaced by a diffuser composed of horizontal plates which serve also as eliminators to prevent water from being carried outward against the air current.

c The humidifying chamber is longer in order to give greater opportunity for saturation.

d The eliminator is not as deep and is not provided with flooding nozzles.

THE DEHUMIDIFIER

12 In the dehumidifier (Figs. 8 and 9) relatively cold spray water is used to condense the moisture out of the air. This water is either refrigerated or taken from an artesian well. When the water is artificially cooled the refrigerating coils are usually placed in a chamber underneath the spray chamber, and the water is so distributed as to flow uniformly over the cold surface, dropping to the tank underneath. The dehumidifier has its sprays opposed to the direction of air flow as in the humidifier, but differs from the latter in having usu-

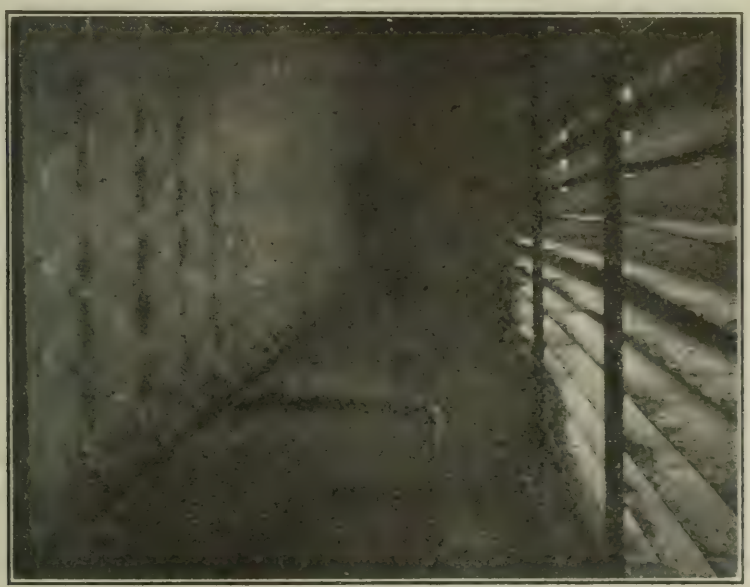


FIG. 7 HUMIDIFIER SHOWING SPRAYS IN OPERATION

ally two sets of sprays in series instead of one. Two or more dehumidifiers are frequently placed in series when the range of air temperature is great or when an economy of cooling water is essential.

ROTARY STRAINER

13 Whenever the air handled by air-conditioning apparatus is full of impurities, such as lint in textile mill applications, the ordinary type of strainer would require too much attention and an automatic, self-cleaning rotary strainer, shown in Fig. 10, is essential. This consists of a fine-mesh brass or copper screen, covering a cylin-

drical drum. As this drum rotates it is cleaned at the water surface by a revolving brush. The use of this strainer is of great advantage in humidifiers, since it permits the use of smaller nozzles which give a more finely divided spray, with increased efficiency.

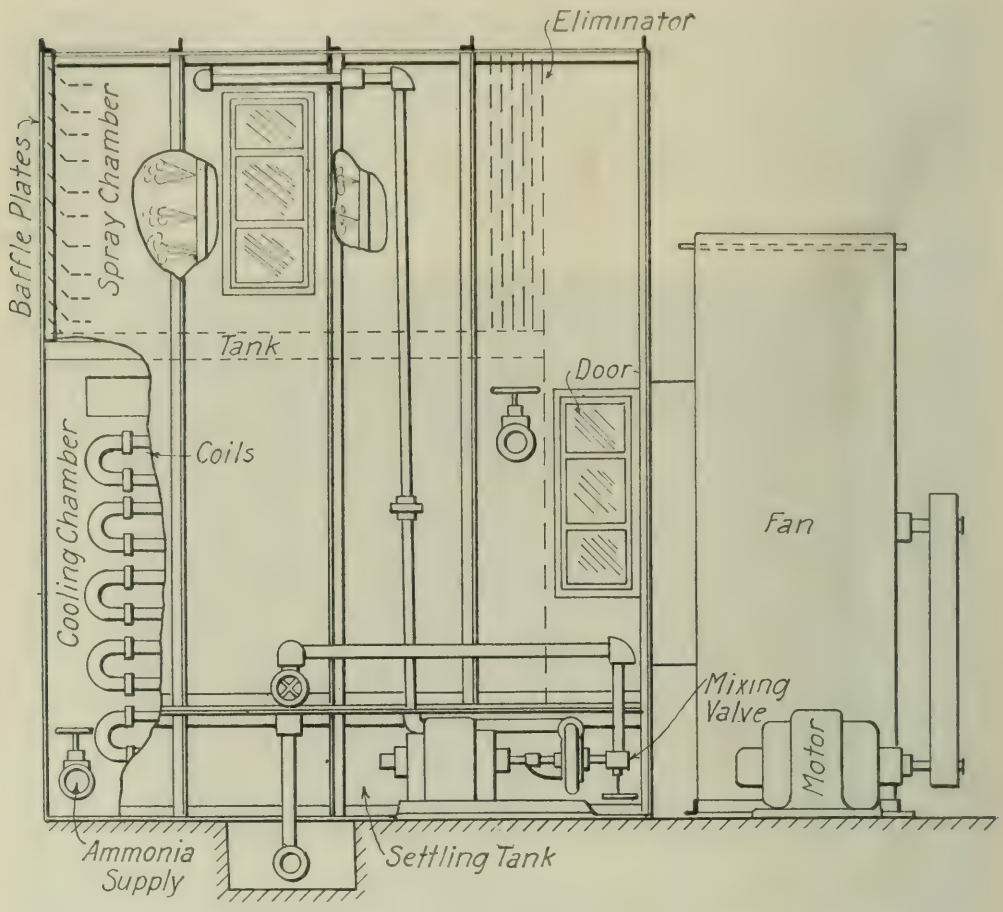


FIG. 8 SECTIONAL VIEW OF DEHUMIDIFIER

DEWPOINT METHOD OF HUMIDITY CONTROL

14 Any one of the three spray types of air conditioners previously described are admirably adapted for humidity control by what is known as the dewpoint method. This system is applicable only where the absolute moisture content of the air in the room is unaffected to any great extent by extraneous sources of moisture supply or by moisture absorption. It depends upon supplying the enclosure with conditioned air having a definite dewpoint and maintaining a predetermined relationship between this dewpoint temperature and the room temperature. The dewpoint of the air supply is deter-

mined by saturating the air and removing all free moisture at the apparatus at a definite temperature. This dewpoint will evidently remain constant regardless of subsequent variations in air temperature. It may be shown that the percentage of relative humidity in an enclosure is dependent upon the difference between the dewpoint temperature and the room temperature and that it is substantially constant for any variation in room temperature so long as the differ-

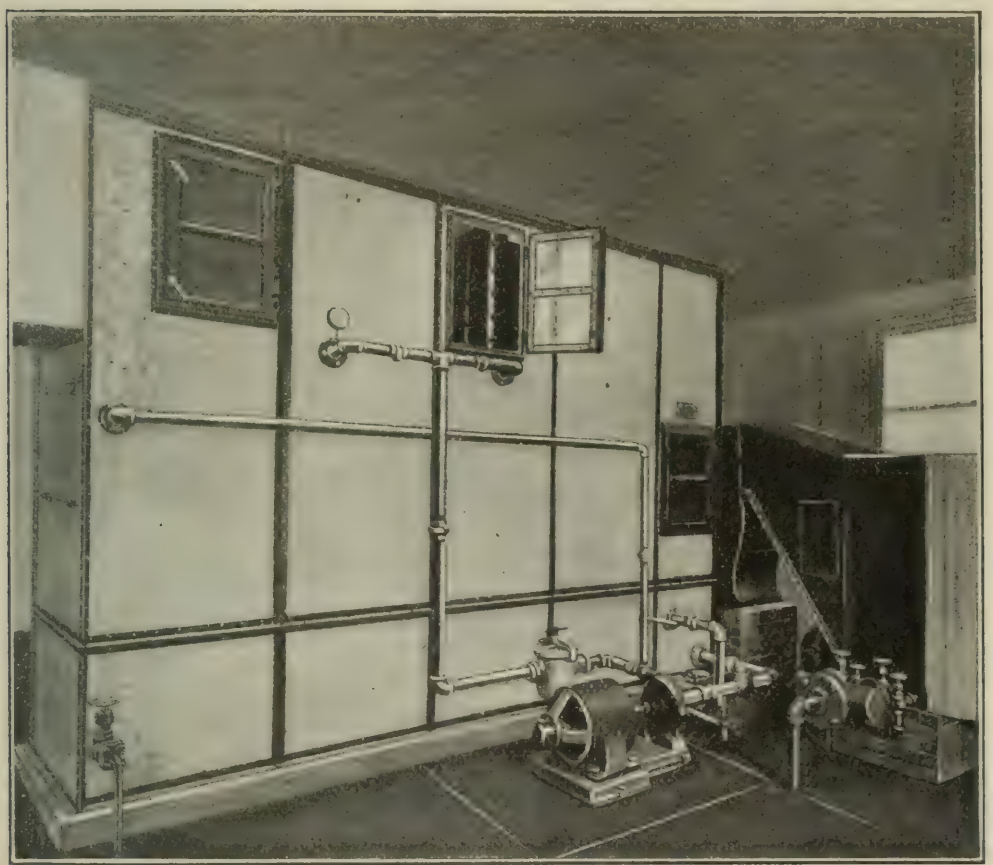


FIG. 9 GENERAL VIEW OF DEHUMIDIFIER

ence between the dewpoint and room temperatures is maintained constant (Tables 1 and 2).

15 It is evident that this system is particularly adapted to thermostatic control of (*a*) the dewpoint (saturation temperature at the apparatus) and the room temperature independently; (*b*) the dewpoint with reference to a variable room temperature; or (*c*) the room temperature with reference to a variable dewpoint temperature. System (*a*) is generally applied to air washers and humidifiers

TABLE 1 DEWPOINT TEMPERATURES AND TEMPERATURE DIFFERENCES REQUIRED IN CARRIER SYSTEM OF HUMIDIFYING FOR VARIOUS PERCENTAGES OF HUMIDITY AND ROOM TEMPERATURES

Room Temperature Deg.	Percentage Relative Humidity																			
	85		80		75		70		65		60		55		50		45		40		35	
	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature	Dewpoint Temperature	Difference Between Dewpoint and Room Temperature
65	60.5	4.5	58.8	6.2	56.75	8.25	54.75	10.25	52.8	12.2	50.7	14.3	48.3	16.7	45.8	19.2	43.0	22.0	40.0	25.0	36.75	28.25
70	65.4	4.6	63.5	6.5	61.6	8.4	59.6	10.4	57.5	12.5	55.3	14.7	53.0	17.0	50.5	19.5	47.5	22.5	44.5	25.5	41.0	29.0
75	70.1	4.9	68.25	6.75	66.3	8.7	64.3	10.75	62.25	12.75	60.0	15.0	57.75	17.25	55.25	19.75	52.75	22.75	49.0	26.0	45.5	29.5
80	74.8	5.2	73.2	6.8	71.2	8.8	69.25	10.75	67.2	12.8	64.8	15.2	62.3	17.7	59.75	20.25	56.75	23.25	53.5	26.5	49.75	30.25
85	79.75	5.25	78.0	7.0	76.2	8.8	74.1	10.9	71.75	13.25	69.4	15.6	66.9	18.1	64.25	20.75	61.25	23.75	58.0	27.0	54.2	30.8
90	84.75	5.25	83.0	7.0	80.9	9.1	78.8	11.2	76.7	13.3	74.2	15.8	71.6	18.4	68.75	21.25	65.75	24.25	68.55	27.5	59.0	31.0
95	89.7	5.3	87.8	7.2	85.75	9.25	83.6	11.4	81.3	13.7	78.8	16.2	76.25	18.75	73.35	21.65	70.3	24.7	67.0	28.0	63.2	31.8

under winter conditions, where the outside temperature is considerably lower than the room temperature and to dehumidifiers where it is possible to maintain a definite dewpoint temperature throughout the entire year. Outlets in the dewpoint air-conditioning are shown in Fig. 11.

16 However during summer conditions the saturation point at the apparatus will frequently and unavoidably be higher than the required minimum dewpoint. Under such variable temperature

TABLE 2 HEAT REQUIRED TO CONDITION 1000 CU. FT. OF AIR (MEASURED AT 70 DEG. FAHR.) FROM VARIOUS ENTERING WET-BULB TEMPERATURES TO VARIOUS DEWPOINT TEMPERATURES

Wet-Bulb Temperature of Entering Air	At 70 Deg. Fahr., 30 Per Cent Humidity, Dewpoint, 37.25 Deg. Fahr.			At 70 Deg. Fahr., 40 Per Cent Humidity, Dewpoint 44.5 Deg. Fahr.			At 70 Deg. Farh. 50 Per Cent Humidity, Dewpoint 50.5 Deg. Fahr.		
	Sensible Heat	Latent Heat	Total Heat	Sensible Heat	Latent Heat	Total Heat	Sensible Heat	Latent Heat	Total Heat
−10	856	338	1194	981	471	1452	1086	567	1653
0	673	311	984	802	444	1246	907	540	1447
10	480	270	750	622	403	1025	730	498	1228
20	307	203	510	443	336	779	550	433	983
30	200	100	300	263	233	496	370	330	700
40				82	96	178	190	194	384

Wet-Bulb Temperature of Entering Air	At 70 Deg. Fahr., 60 Per Cent Humidity, Dewpoint 55.3 Deg. Fahr.			At 70 Deg. Fahr., 70 Per Cent Humidity, Dewpoint 59.6 Deg. Fahr.			At 70 Deg. Fahr., 80 Per Cent Humidity, Dewpoint 63.5 Deg. Fahr.		
	Sensible Heat	Latent Heat	Total Heat	Sensible Heat	Latent Heat	Total Heat	Sensible Heat	Latent Heat	Total Heat
−10	1161	699	1860	1243	801	2044	1310	935	2245
0	991	672	1663	1066	774	1840	1131	908	2039
10	814	631	1445	888	733	1621	955	867	1822
20	635	565	1200	710	667	1377	779	802	1581
30	457	463	920	532	565	1097	600	700	1300
40	276	327	603	353	430	783	423	564	987
50	83	137	220	154	240	394	244	375	619
60							63	118	181

conditions it is necessary to control temperature with reference to the dewpoint according to system (*c*), and a humidifier is employed to give the air complete saturation under these conditions. One of the forms of differential thermostats which will be described later effects this control.

AUTOMATIC HUMIDITY CONTROL

17 In many industrial installations where humidifying or dehumidifying systems are used, some means of positively and accurately maintaining the proper temperatures and humidities is essential. While much can be accomplished by hand regulation, this would require the constant attention of a highly skilled operator, which in most instances is impracticable. In many processes of manufacturing, as, for example, the weaving of silk and in the conditioning of tobacco for the manufacture of cigars, a uniformity of humidity

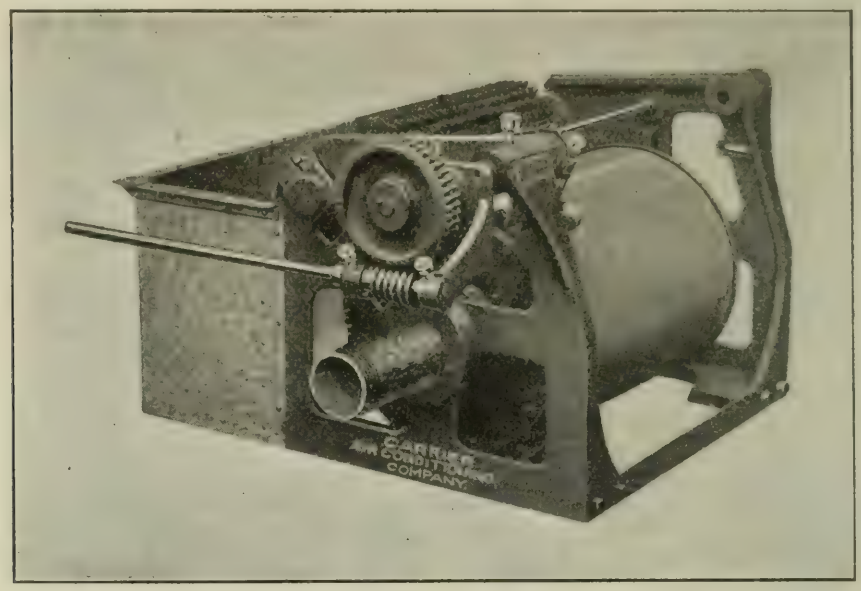


FIG. 10 ROTARY STRAINER

conditions is quite as essential as the quantity of moisture, as any variation in humidity, either above or below a standard, reduces the output and causes lack of uniformity in the product. In many cases a sensitive automatic humidity control is as important as some means of humidifying. There are three distinct methods by which such automatic control can be secured:

a By two separate thermostats, one of which, *D*, is placed at the humidifier just behind the eliminator plates shown at *A* (Fig. 2). This controls the temperature of the dewpoint, by a valve or damper operating automatically, governing a means of varying the temperature of the spray water, of the entering air, or of both in conjunction. The other thermostat, placed in the

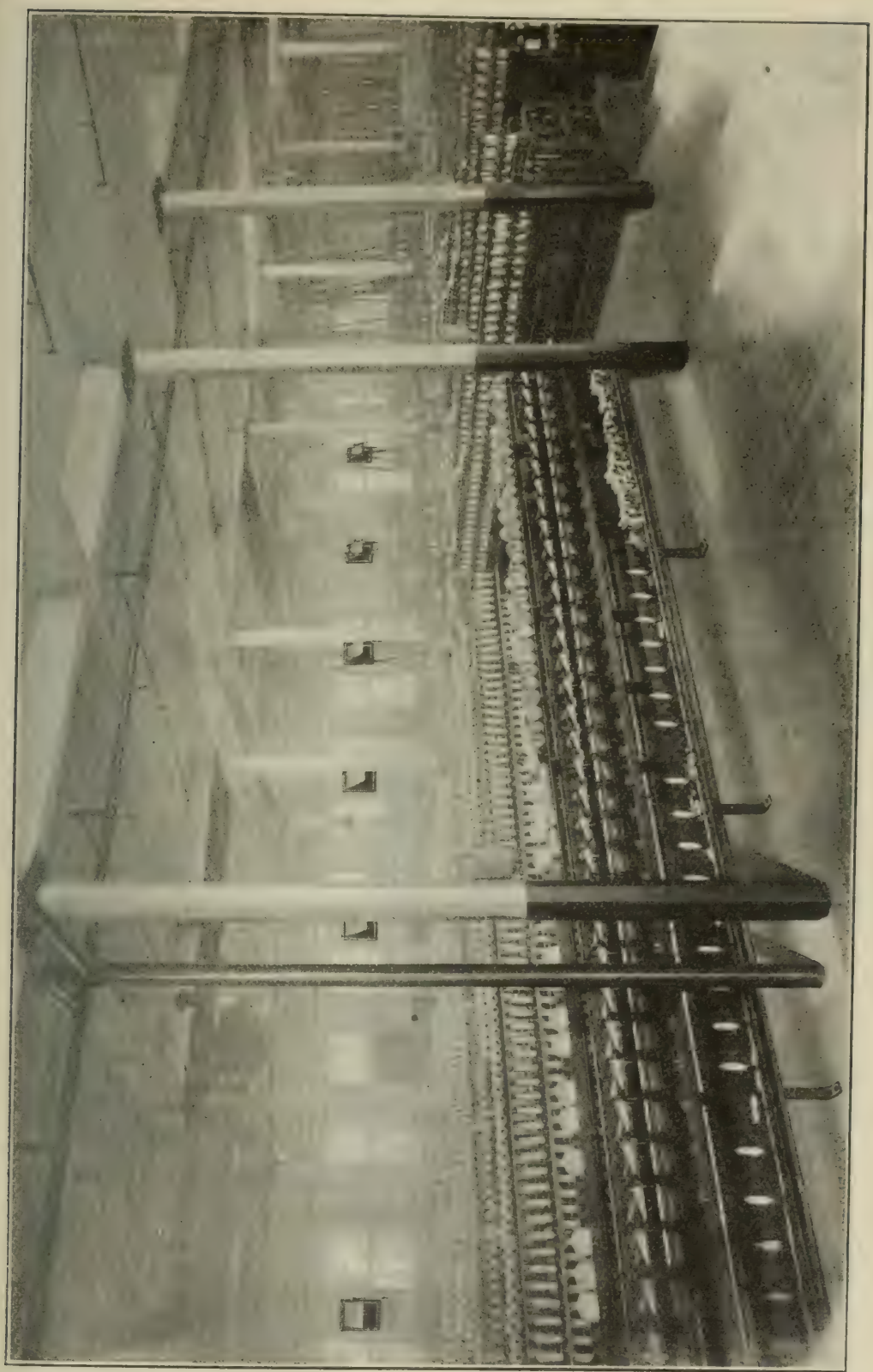


FIG 11 SPINNING ROOM SHOWING OUTLETS IN DEWPOINT AIR-CONDITIONING SYSTEMS.

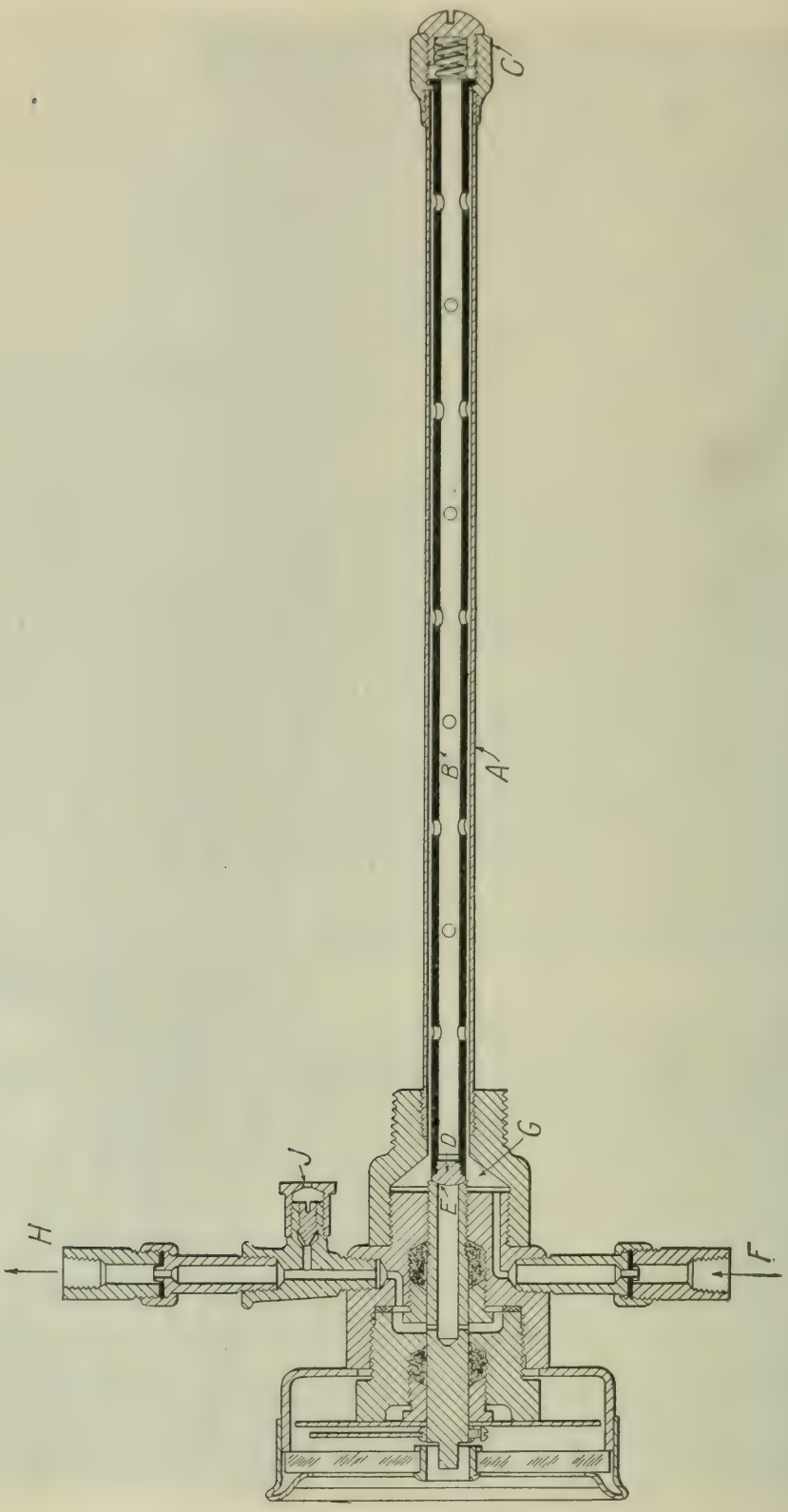


FIG. 12 DEWPOINT THERMOSTAT

room where the humidity is controlled, maintains a constant room temperature, either by controlling the temperature of the air entering the room, or by controlling some source of heat within the room. With these two temperatures maintained constant, the percentage of humidity in the room will remain constant, and will depend upon the difference between the dewpoint temperature maintained at the humidifier and the temperature maintained in the room, as previously shown in Table 1.

b By a differential thermostat either of the form shown in Fig. 12, or as shown in Fig. 14. This type of dewpoint control is required wherever it is impracticable to maintain either a constant dewpoint or a constant room temperature. In this method there are two elements, one of which is exposed to the dewpoint temperature, while the other is exposed to the room temperature. They are so connected that they act conjointly upon a single thermostatic valve connected with operating motors arranged to control the dewpoint temperature in relation to the variable room temperature, or to control the room temperature with respect to the variable dewpoint temperature.

c By means of some form of differential hygrostat, as shown in Figs. 17, 21 and 22. This controls the wet-bulb temperature with respect to the dry-bulb temperature, so as to maintain a constant relative humidity without regard to the dewpoint or variation in room temperature.

DEWPOINT THERMOSTAT

18 The type of thermostat usually employed in maintaining a constant dewpoint is shown in Fig. 12. This consists of an outer expansive member *A*, usually brass, and an inner non-expansive member *B* of nickel steel. These two members are firmly connected at the end *C*. The other end of the inner member *B* is provided with a bronze valve *D*, ground to fit the adjustable valve seat *E*, supported by the member *A*. Compressed air is admitted through the connection *F* to the annular chamber *G* between the inner and outer tubes. As the outer member expands, the valve *D* recedes from its seat, allowing the compressed air to escape into the outlet connection *H*, which connects with the diaphragm valve controlling the

temperature of the spray water, so as to reduce the temperature of the dewpoint. When the dewpoint temperature falls below the point desired, the outer member contracts, closing off the air supply to the

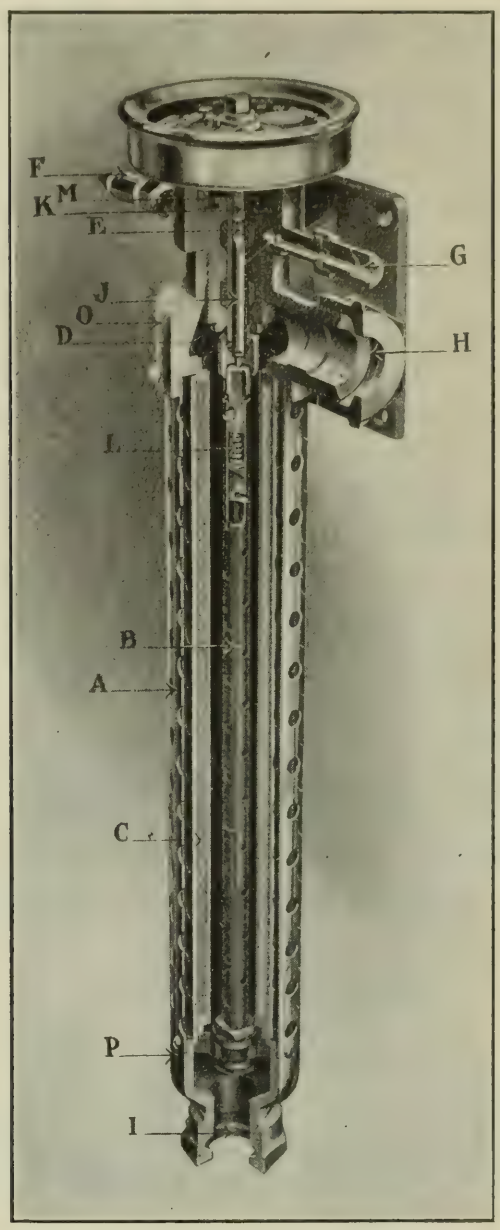


FIG. 13 METALLIC DIFFERENTIAL THERMOSTAT

diaphragm valve, connected to *H*, and the air pressure to the diaphragm motor is released through the adjustable vent *J*. This vent allows an air leak varying with the pressure on the diaphragm

motor. Therefore the relation of the area of opening through the valve *D-E*, to the constant area of the vent opening *J*, determines

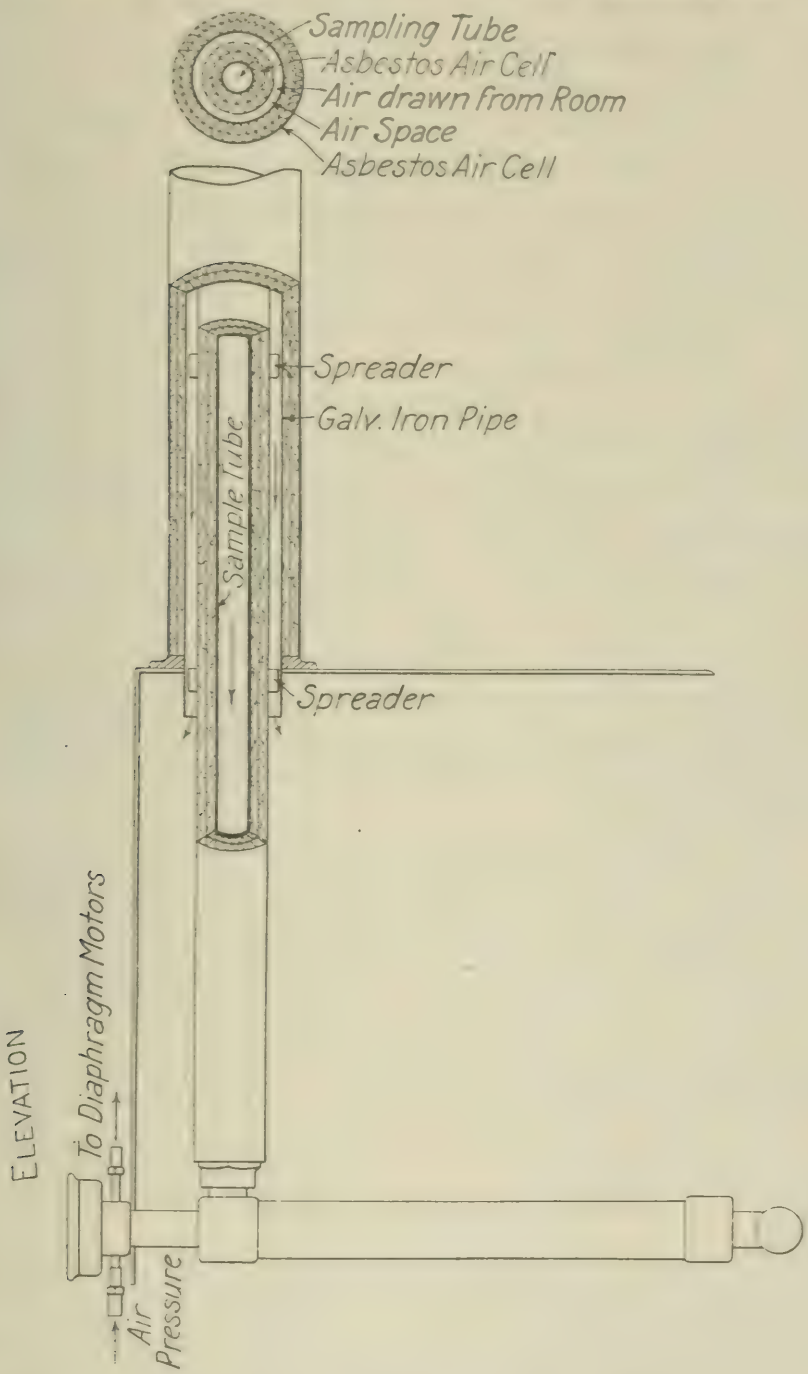


FIG. 14 SAMPLING TUBE OF DIFFERENTIAL THERMOSTAT

the graduated pressure on the diaphragm motor, at any instant. This is found in practice to give a very sensitive as well as positive control.

METALLIC DIFFERENTIAL THERMOSTAT

19 The differential thermostat shown in Fig. 13, resembles the dewpoint thermostat in many features of construction, except that it consists of two expansible members operating conjointly, instead of one. The outer member *A* is subjected to the dewpoint temperature at the humidifier, while the inner member *B* is subjected to a strong current of air drawn from the room to be conditioned by means of an aspirator or fan, through an insulated tube. The two members are insulated from each other by an annular space *C*, filled with mineral wool. These two members are connected at the base *P* and act conjointly upon the double-ported valve, *D-E*. Compressed air is admitted to the chamber *M* through the connection *F*, and passes through the valve *E* to the chamber *J*, which is joined by the connection *G* to the diaphragm motor closing off the supply of heat to the spray water. Whenever the difference between the dewpoint and the room temperature is greater than that for which the instrument is adjusted, by means of the dial, the outer member *A* contracts with reference to the inner member *B*. This closes the port *E*, cutting off from the chamber *J* the supply of compressed air, simultaneously opening the port *D*, and allowing the air to escape from the diaphragm valve, which then operates to raise the temperature of the spray water.

20 The means employed for bringing the sample of air from the room to the thermostat is shown in Fig. 14. The sample of air is drawn through the insulated inner tube, while room air is also drawn through the annular space between this insulation and the outer pipe by means of a fan draft. This outer tube is also insulated outside of the room. The sampling tube usually connects with the thermostat at *H*, while the aspirator is connected at *I*.

FLUID DIFFERENTIAL THERMOSTAT

21 This type of differential thermostat (Fig. 15) is adapted for use where there are several floors to be controlled independently and conditioned from a central apparatus. The dewpoint member and room member each consist of hermetically sealed chambers filled with air or other fluid under pressure. The air or fluid in either of these members will tend to expand and will increase the pressure, with constant volume, in direct proportion to the increase of temperature. The pressure in the dewpoint member, *B*, is conveyed either directly

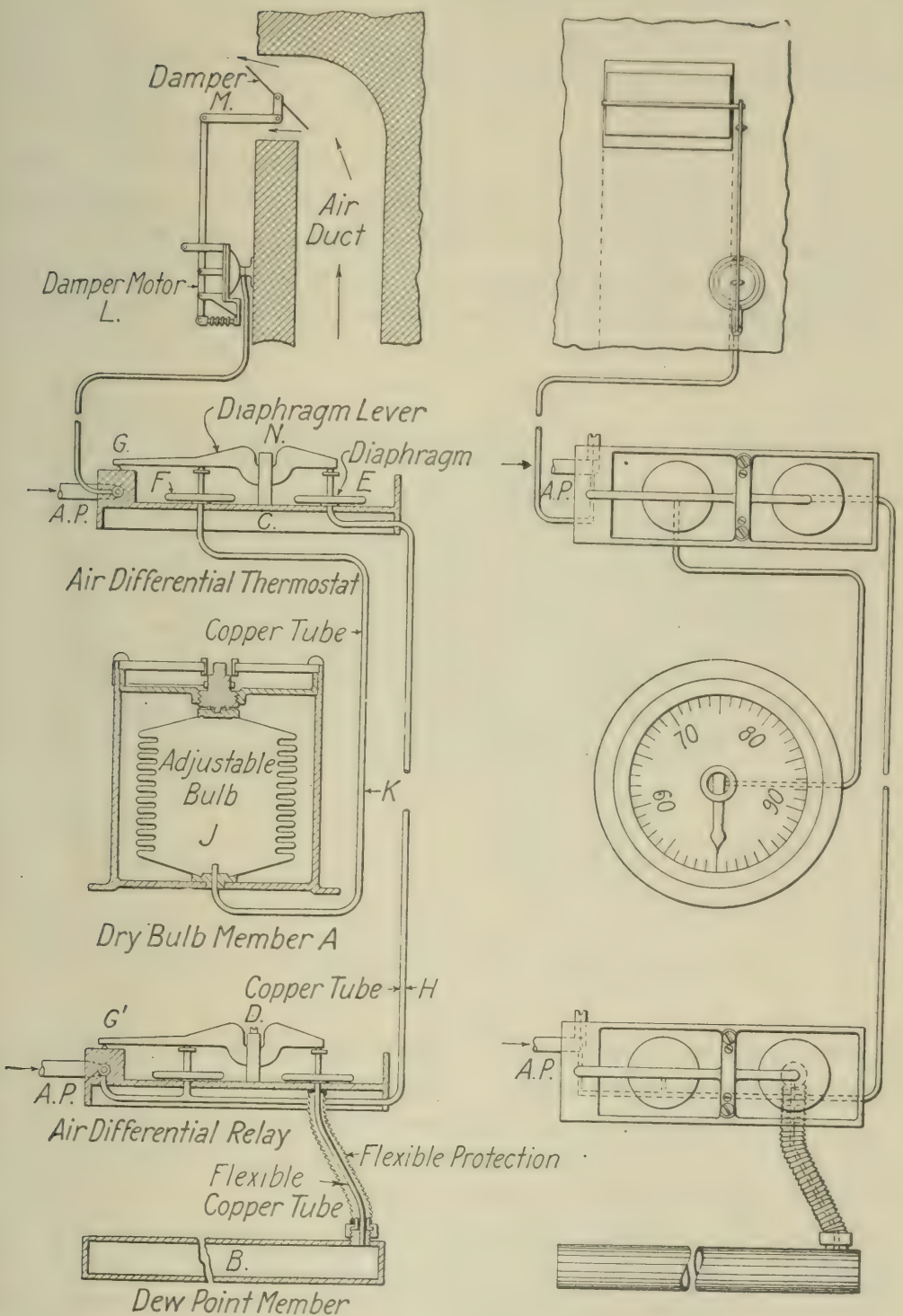


FIG. 15 FLUID DIFFERENTIAL THERMOSTAT AND HUMIDITY CONTROL

or proportionately to the diaphragm E of the differential thermostat C, through the differential relay D and connecting tube H. The room member A, which has an adjustable bulb J, connects through tube K to diaphragm F of the differential thermostat. The pressures or proportionate pressures in these two elements are thus opposed to each other through the lever N. Any unbalancing of pressures operates the valve G, which is shown in detail in Fig. 16. So long as the temperature difference between the dewpoint temperature at the apparatus and the room temperature remains constant, the pressures on the diaphragms E and F will remain balanced and no movement of

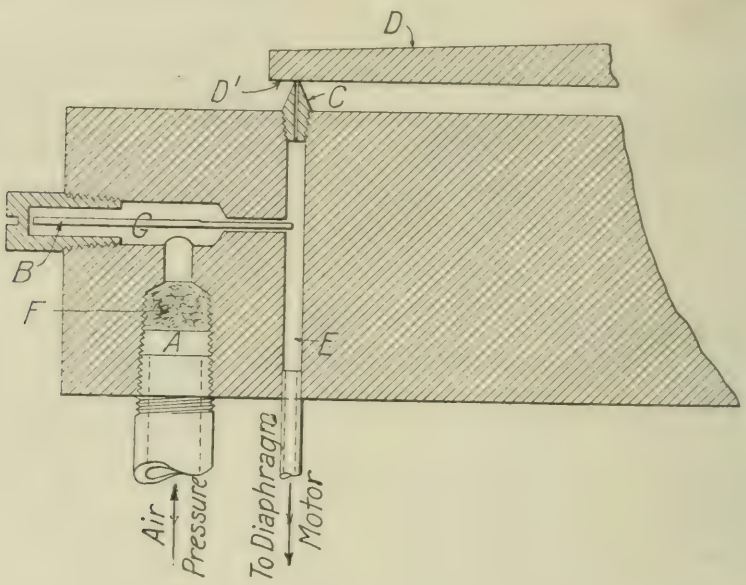


FIG. 16 CONSTRUCTION OF THERMOSTATIC VALVE

the valve G will take place. If, however, an excess of moist air is admitted to the room, until its temperature is reduced to a point where the difference between it and the dewpoint temperature is less than the predetermined amount, the pressure on the diaphragm F will be less than the pressure on diaphragm E. This will permit the valve G to operate, bringing pressure on damper motor L and closing the damper M, cutting off the supply of cool moist air.

22 Adjustable bulb J, exposed to the room temperature and provided with a dial, permits the thermostat to be adjusted for any desired difference between dewpoint temperature and room temperature, and consequently for any percentage of relative humidity regardless of variation in room temperature.

23 The construction of the thermostatic valves G and G' is shown in Fig. 16. Compressed air is admitted at A through filter F into the restriction chamber G, and also in a constant amount through restriction into chamber E, connecting in case of the differential

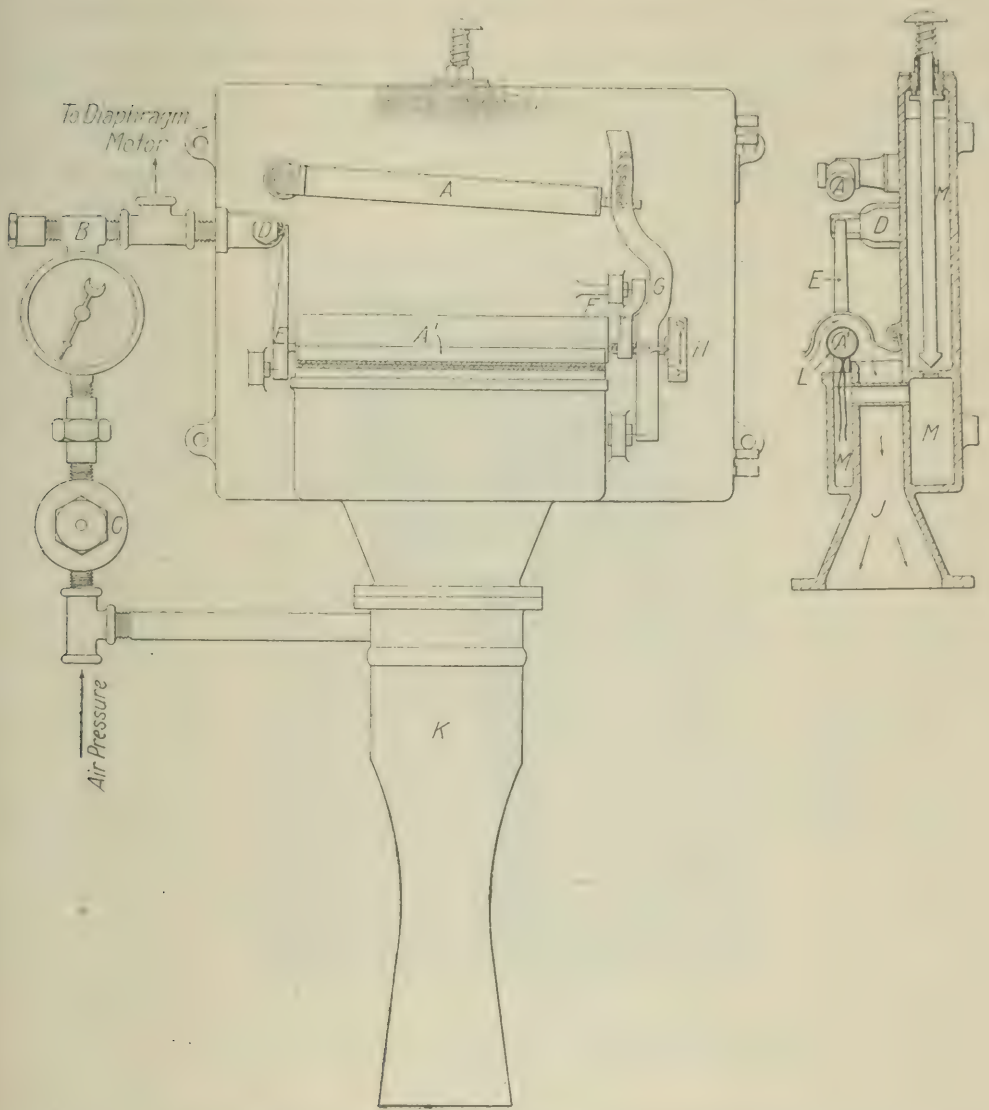


FIG. 17 CARRIER DIFFERENTIAL HYGROSTAT

relay with diaphragm E of the differential thermostat, and in case of the differential thermostat with the diaphragm motor controlling damper M. Upon the lower side of the diaphragm lever D is a uniformly ground and polished surface D' which approaches or recedes from the nozzle C as the pressure on one diaphragm overcomes that on the other, increasing or decreasing the pressure in E correspondingly.

DIFFERENTIAL HYGROSTAT

24 In many instances the dewpoint system of humidity control cannot be applied to advantage. In such cases a differential hygrostat may be employed. The differential hygrostat (Figs. 17 and 18) consists of two members, one of which is subjected to the dry-bulb temperature of the room, while the other is subjected to the wet-bulb temperature. The expansive dry-bulb member A, and the wet-bulb member A', of hard rubber tube, connected by levers F and G, operate conjointly the valve lever E. Valve D and restriction B are

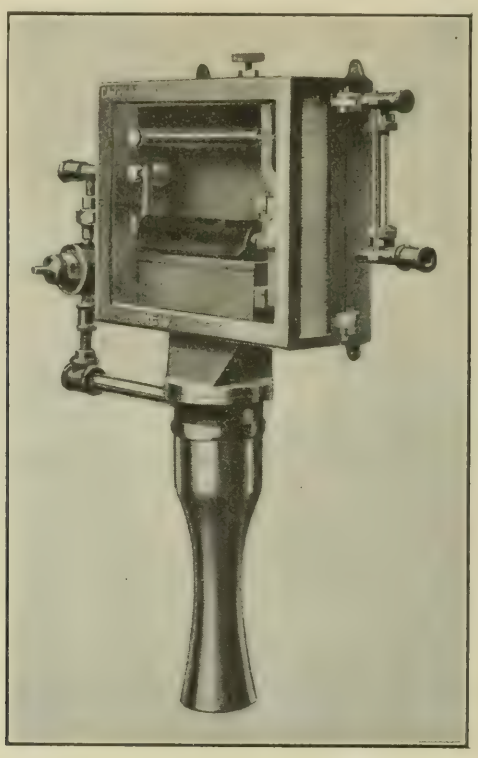
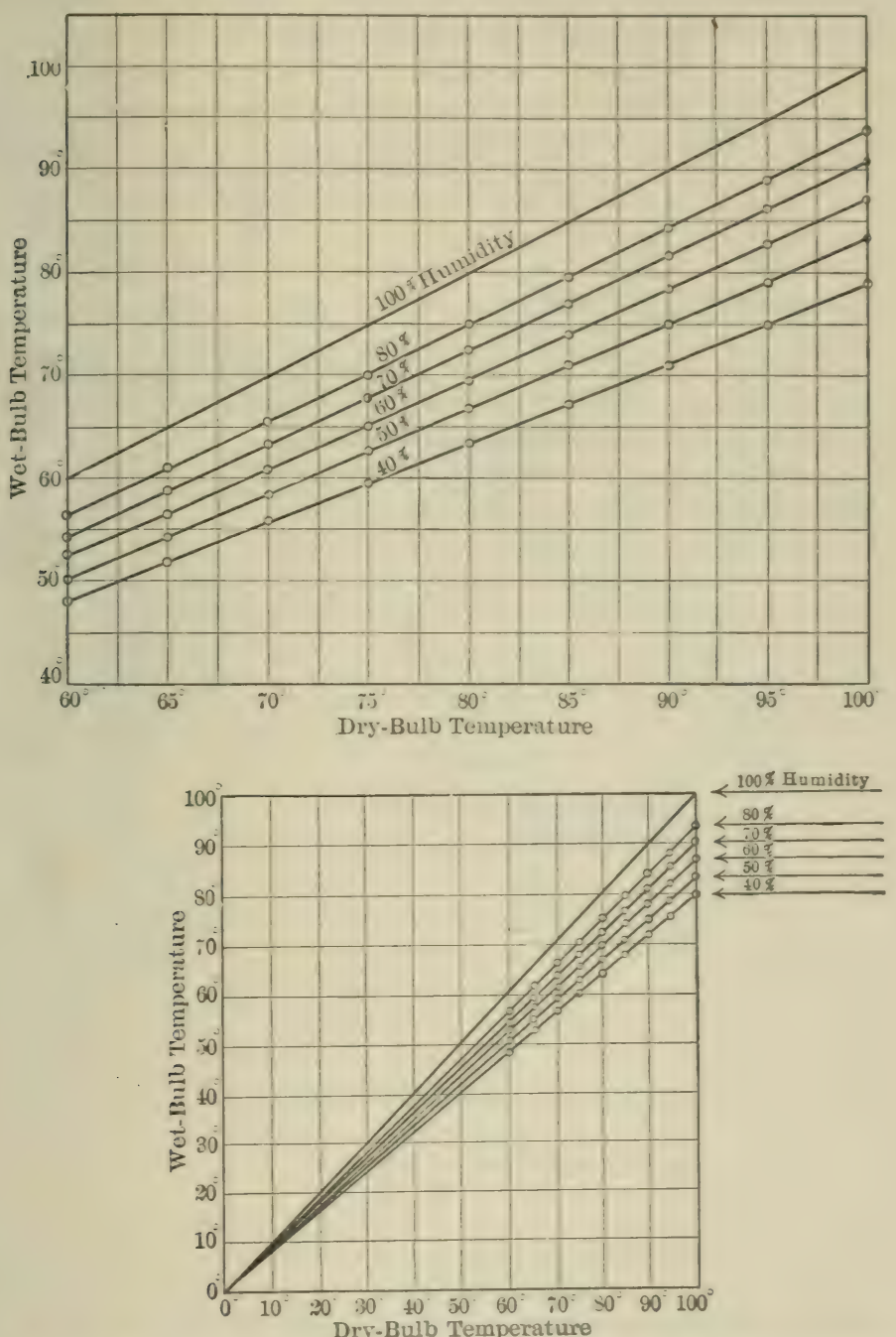


FIG. 18 CARRIER DIFFERENTIAL HYGROSTAT

similar in construction to the valve shown in Fig. 16 and operate in the same way. As will be seen from Figs. 19 and 20, the difference between the dry and wet-bulb temperature for a given per cent of humidity is not constant at different dry-bulb temperatures, i.e., a constant difference maintained between dry and wet-bulb temperatures would not result in a constant per cent of humidity during this change. For instance, with 50 per cent of humidity at a dry-bulb temperature of 70 deg., the wet-bulb temperature is 58.42 deg., a difference of 11.58 deg. At 80 deg. the wet-bulb temperature is 66.65 deg.,

or a difference of 13.35 deg. Thus it will be seen that the dry-bulb temperature has risen 10 deg., while the wet-bulb has risen but 8.23 deg.



FIGS. 19 AND 20 RELATION BETWEEN WET AND DRY-BULB TEMPERATURE FOR DIFFERENT RELATIVE HUMIDITIES

a difference of 1.77 deg. To compensate for this difference a change of leverage is effected, as indicated by graduations on lever *G*. An

adjustment H permits of regulation to any desired humidity. As shown in sectional view, a wick L covers A', being moistened by water held in chamber M. A sufficient current of room air is drawn by means of an aspirator K, over A' through chamber J.

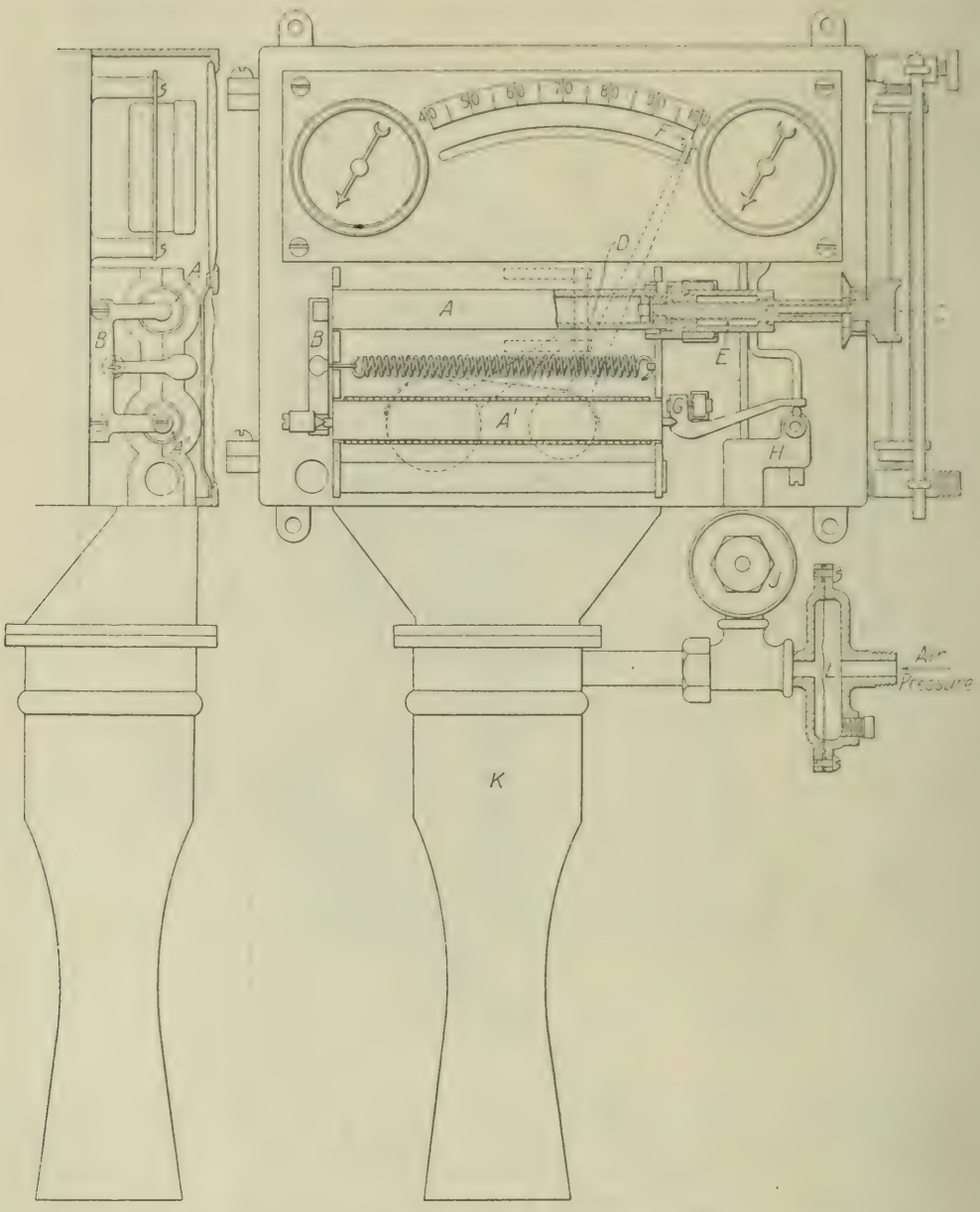


FIG. 21 IMPROVED FORM OF CARRIER DIFFERENTIAL HYGROSTAT

25 An improved form is shown in Fig. 21. It differs essentially in three features: (*a*) the air current is drawn over both expansive members; (*b*) the action of a portion of the expansive member, A, is cut out in the adjustment instead of its action being decreased by a

change of leverage; (*c*) the unique feature of this form is that it requires but one adjustment. A differential thread is used, making it a compound adjustment. The pitch of the thread *D*, in *A*, is greater than the pitch of the thread on the adjusting screw. As the screw moves along to shorten *A*, sufficient to compensate for change of difference of dry and wet-bulb temperatures for varying dry-bulb temperature, as shown by graduations *A* and *A′* acting conjointly

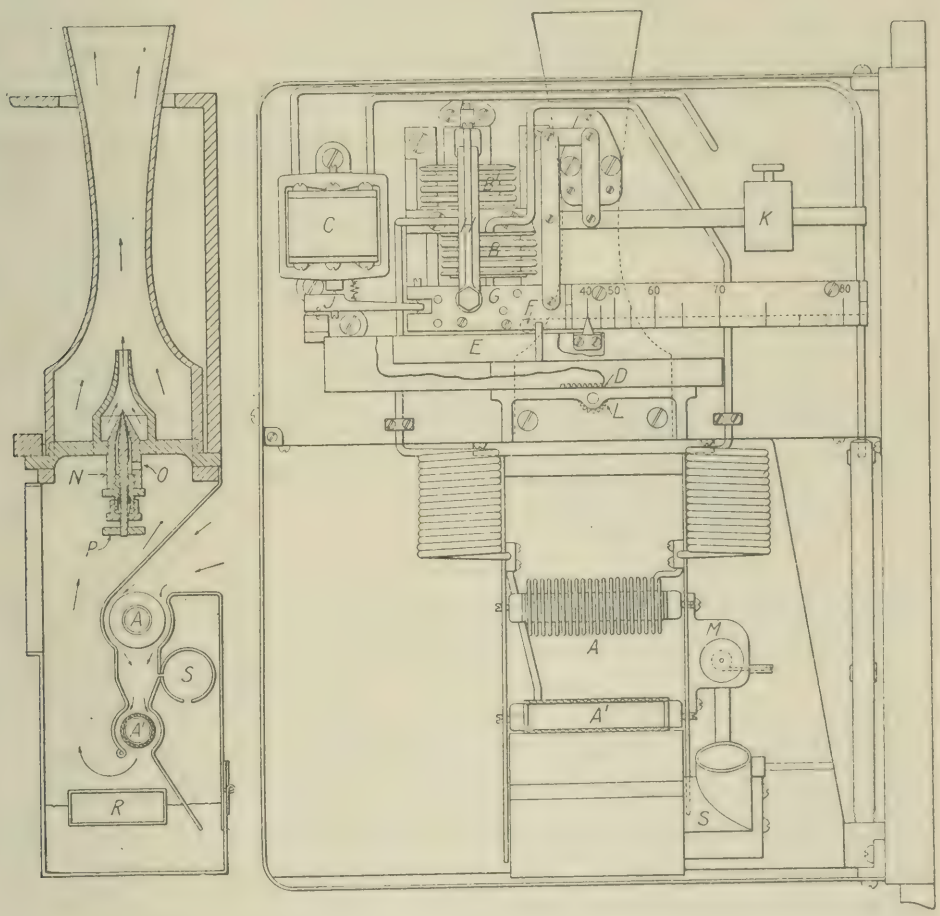


FIG. 22 VAPOR PRESSURE HYGROSTAT

through rocker arm *B* are carried back, so adjusting lever *G*, as to maintain the humidity for which the indicator is set.

26 *Vapor Pressure Hygrostat.* Fig. 22 shows an improved form of hygrostat operated by the relative pressures of a volatile liquid subjected to the wet and dry-bulb temperatures. These vapor pressures act through suitable diaphragms upon a common lever at variable distances from the fulcrum. Referring to Fig. 22, *A* is the dry-bulb member, *A′* the wet-bulb member, covered with a

wick or absorbent material and moistened by atomized spray produced by the atomizing nozzle M (Fig. 22, shown in detail in Fig. 23). A current of air is drawn over both the dry and wet bulbs by means of the aspirator, as shown in sectional view (Fig. 22). The vapor pressures in these bulbs are conducted to the corresponding diaphragms by means of tubes. Diaphragm B' is connected with the wet bulb A', and the diaphragm B connects with the dry bulb A. The diaphragm B' acts upward on the lever G through the link H, while the diaphragm B acts downward at point G, between the link H and the fulcrum F. Fulcrum F, consists of a hardened steel ball having a rolling contact with the lever G, and the adjusting bar E. The adjusting bar E is moved by a rack and pinion D-L. The scale in per cent of humidity is attached to the lever G, while the pointer is carried on the adjusting bar E and therefore moves proportionally with the fulcrum. Any

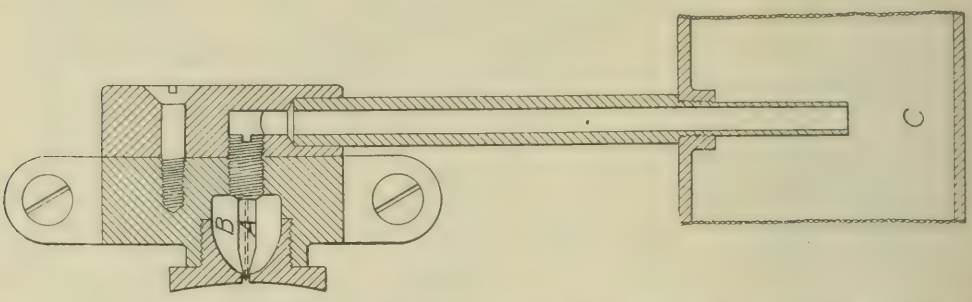


FIG. 23 CONSTRUCTION OF ATOMIZER USED IN CARRIER VAPOR PRESSURE HYGROSTAT

adjustment of F changes the relative leverage of B and B'; so that by adjustment of it any ratio of leverage may be secured. Any movement of the lever G operates the balanced thermostatic valve C, which controls the source of humidification. The particular fluid adapted for use in this type of instrument is sulphur dioxide, the temperature pressure properties of which are remarkably adapted to this purpose, as shown in Table 3 and Fig. 24. The ratio of sulphur dioxide pressures corresponding to wet and dry-bulb temperatures is substantially constant at any per cent of humidity for any range of dry-bulb temperatures between 60 and 100 deg. This makes it possible to oppose the pressures at any desired ratio corresponding to the required per cent of humidity, where it will control throughout any range of dry-bulb temperatures.

27 *Recording Hygrometer.* The same principle is applied to great advantage in the recording hygrometer shown in Fig. 25. In this we have dry bulb A and wet bulb A' connected respectively to dia-

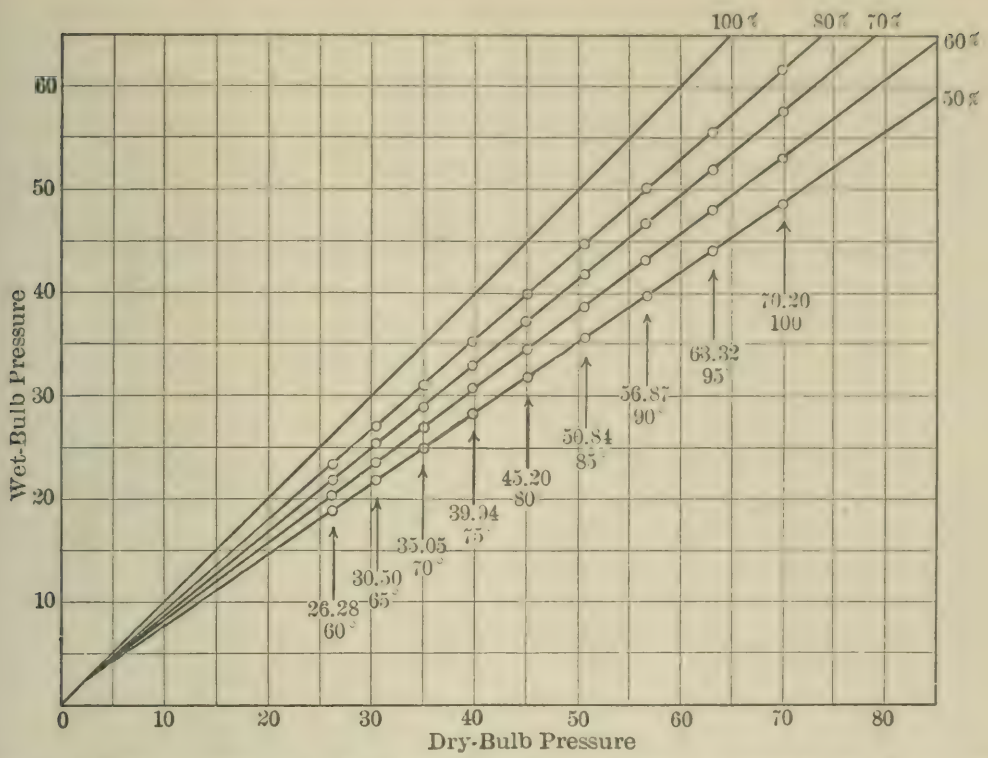


FIG. 24 PSYCHROMETRIC WET AND DRY-BULB TEMPERATURES FOR VARIOUS PERCENTAGES OF HUMIDITY

TABLE 3 RELATIVE PRESSURES OF SULPHUR DIOXIDE VAPOR CORRESPONDING TO PSYCHROMETRIC WET AND DRY-BULB TEMPERATURES FOR VARIOUS PERCENTAGES OF HUMIDITY

Dry-Bulb Temperature, Deg. Fahr.	50 Per Cent				60 Per Cent			70 Per Cent			80 Per Cent		
	SO_2 Pressure (Gage)	Wet-Bulb Temperature, Deg. Fahr.	SO_2 Pressure (Gage)	Ratio of Pressures —2.75 Lb.	Wet-Bulb Temperature, Deg. Fahr.	SO_2 Pressure (Gage)	Ratio of Pressures —2.75 Lb.	Wet-Bulb Temperature, Deg. Fahr.	SO_2 Pressure (Gage)	Ratio of Pressures —2.75 Lb.	Wet-Bulb Temperature, Deg. Fahr.	SO_2 Pressure (Gage)	Ratio of Pressures —2.75 Lb
60	26.28	50.24	18.90	0.6863	52.28	20.35	0.7479	54.30	21.80	0.8096	56.28	23.30	0.8733
65	30.50	54.32	21.85	0.6863	56.54	23.55	0.7495	58.81	25.35	0.8144	61.05	27.10	0.8775
70	35.05	58.42	24.95	0.6873	60.86	26.95	0.7489	63.22	28.95	0.8111	65.63	31.05	0.8761
75	39.94	62.53	28.35	0.6883	65.22	30.65	0.7502	67.80	33.00	0.8134	70.34	35.35	0.8766
80	45.20	66.65	31.95	0.6878	69.58	34.60	0.7503	72.39	37.35	0.8151	75.07	39.95	0.8762
85	50.84	70.72	35.75	0.6862	73.95	38.85	0.7507	77.00	41.95	0.8151	79.72	44.85	0.8754
90	56.87	74.89	39.80	0.6846	78.32	43.35	7.7502	81.61	46.90	0.8158	84.50	50.15	0.8758
95	63.32	79.01	44.15	0.6835	82.72	48.20	0.0503	86.04	52.05	0.8139	89.21	55.85	0.8767
100	70.20	83.33	48.80	0.6827	87.07	53.20	0.7479	90.64	57.65	0.8139	94.00	61.95	0.8777

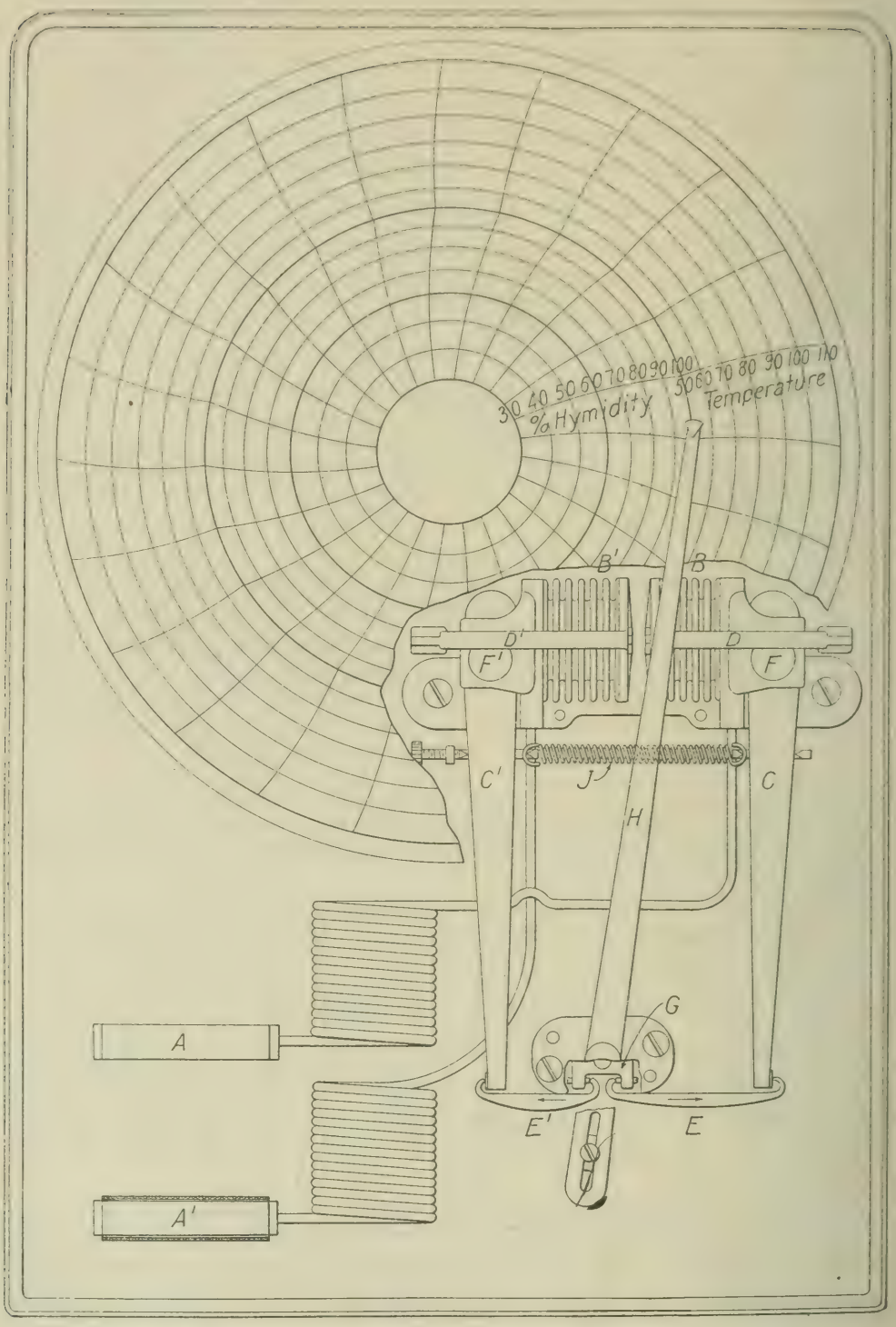


FIG. 25 MECHANISM OF CARRIER RECORDING HYGROMETER

phragms B and B', which act through the links, D and D', upon the levers C and C', which act upon ball-bearing fulcrums F and F'. The relative forces are transmitted from the levers C and C' through the links E and E' to the rocker arm G, which is firmly supported by minute ball bearings. To the rocker arm G, is connected the recording pen H. By referring to Table 3 and Fig. 24, it will be seen that any per cent of humidity may be represented at all temperatures by

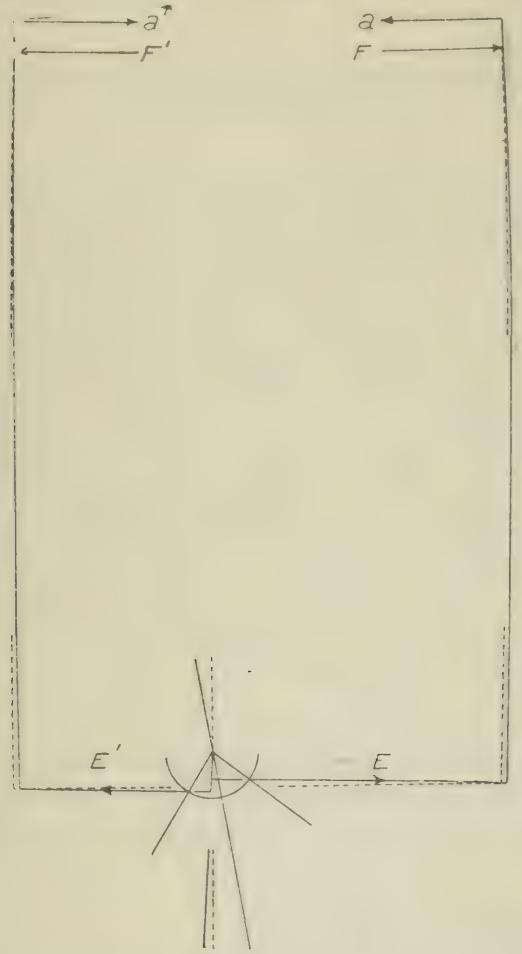


FIG. 26 LEVERAGE DIAGRAM OF CARRIER RECORDING HYGROMETER

a pressure ratio. The rocker arm G acts as an evener. When it is acted upon by the stronger force of the dry bulb through the link E, it rotates toward C, decreasing the everage of E and increasing the leverage of E', until the two moments are exactly balanced as shown in diagram Fig. 26. The angular deflection of the rocker arm will then indicate the ratio of pressures in the diaphragms B and B'. Therefore the angular deflection is constant for any percentage of

humidity regardless of temperature variations. The spring J has an important function as a compensator, both for the small pressure subtractions required to obtain an exact ratio, and for the slight resistance of the diaphragms B and B'. The latter compensation while exceedingly simple, is effectual and complete, allowing for any amount of diaphragm resistance. This permits the instrument to give perfect indications regardless of the temperature variations, or the weight of the material used in the construction of the diaphragms.

ELEMENTS OF DESIGN OF HUMIDIFIERS AND OF AIR-CONDITIONING SYSTEMS

28 The degree of saturation of the air leaving any type of air washer depends upon the intimacy of the contact of the air and water, and upon the relation of the water temperature to the wet-bulb temperature of the entering air. It also depends to some degree upon the length of the spray chamber as well as upon the velocity of the air passing through it. With the centrifugal type of spray nozzles the water pressure is a most important element affecting the degree of saturation. Fig. 27 shows the humidifying effect secured with various velocities and at different pressures on the spray nozzle, in a standard humidifier having four $\frac{3}{32}$-in. orifice centrifugal spray nozzles per sq. ft. These data were obtained from a test in which the wet-bulb depression of the entering air was maintained constant at 16 deg. It will be noted that an increase of $2\frac{1}{2}$ lb. per sq. in. in the spray pressure permitted a greatly increased velocity with perfect saturation, an effect which was undoubtedly due to the increased fineness of the spray rather than to the increase in the amount of water discharged. In this test, as in all standard humidifiers, the water was discharged in the direction opposite the air flow, which greatly increased the efficiency of saturation.

29 When the spray water is recirculated without heating as in warm weather, it remains at all times substantially at the wet-bulb temperature of the entering air while the wet-bulb temperature of the air leaving the washer or humidifier is unchanged. Therefore it follows, in conformance with the theory, that when the air is completely saturated as in the humidifier the air is cooled to the wet-bulb temperature of the incoming air. This cooling effect is due to the transformation of sensible heat into latent heat of evaporation and is therefore in direct proportion to the moisture added to the air. The wet-bulb depression in atmospheric air averages

from 12 to 15 deg. in summer, while occasionally a depression of 20 to 30 deg. is found in extremely hot and dry weather. In every case the humidifier will cool the incoming air a corresponding number of degrees.

30 When saturation is incomplete, as in the ordinary air washer, the wet-bulb depression of the air leaving the washer is found to be a constant percentage of the initial wet-bulb depression, when the air velocity remains constant.

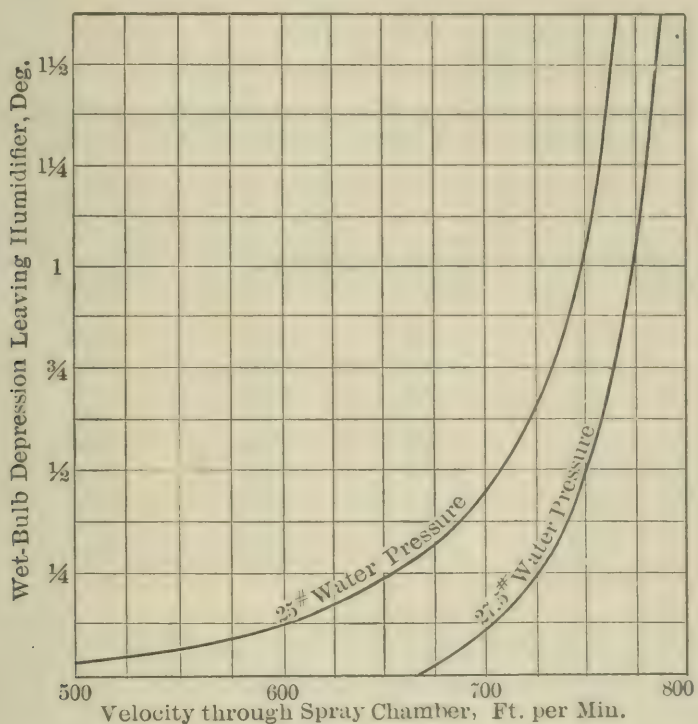


FIG. 27 EFFECT OF SPRAY PRESSURE AND AIR VELOCITY UPON THE DEGREE OF SATURATION. INITIAL WET-BULB DEPRESSION = 16 DEG. FAHR.

31 It also follows that the cooling effect is a constant percentage of the initial wet-bulb depression. This may be expressed by the formulae

$$\frac{t_2 - t'}{t_1 - t'} = R \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots [1]$$

$$\frac{t_1 - t_2}{t_1 - t'} = 1 - R = E \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

where

t' = constant wet-bulb temperature

t_1 = temperature of air entering washer

t_2 = temperature of air leaving washer

R = constant ratio depending upon intimacy of contact, air velocity, etc.

$1 - R = E$ = efficiency of saturation

POWER REQUIRED FOR OPERATING HUMIDIFIERS

32 Table 4 exhibits the power required to saturate 1000 cu. ft. of air per minute at various velocities. This is based on overcoming the resistance of the humidifier, using a fan with a static efficiency of 45 per cent, a fair value.

TABLE 4 RESISTANCE OF CARRIER HUMIDIFIERS AND HORSEPOWER REQUIRED TO HUMIDIFY 1000 CU. FT. OF AIR

Velocity through Spray Chamber in Ft. per Min.	Resistance in In. of Water	Resistance in Oz. per Sq. In.	Horsepower to Move 1000 Cu. Ft. Air per Min. At 45 Per Cent Fan Efficiency	Horsepower for Spray per 1000 Cu. Ft. of Air ($\frac{1}{8}$ Orifice Nozzle)	Total Horsepower Required per 1000 Cu. Ft. of Air
350	0.112	0.0647	0.0391	0.1408	0.1799
400	0.147	0.0850	0.0513	0.1231	0.1744
450	0.186	0.1075	0.0652	0.1095	0.1747
500	0.229	0.1322	0.0800	0.0985	0.1785
550	0.277	0.1600	0.0968	0.0897	0.1865
600	0.330	0.1906	0.1150	0.0822	0.1972
650	0.387	0.2240	0.1350	0.0758	0.2108
700	0.450	0.2600	0.1570	0.0704	0.2274
750	0.516	0.2990	0.1810	0.0658	0.2468

33 This does not include the power required to overcome the resistance of the ducts, which varies considerably, but which should not exceed that required for the humidifier. The resistance of the heating coils is not considered, because in summer when the largest supply of air is usually required the air is by-passed around the heaters, while in winter the requirements are so much smaller that the total horsepower is greatly reduced and the total resistance is but slightly increased.

34 The power required to pump the water is based on the use of centrifugal pumps giving 55 per cent efficiency and using $\frac{1}{16}$-in. orifice nozzles with rotary self-cleaning strainers.

RELATION OF COOLING EFFECT TO PERCENTAGE OF RELATIVE HUMIDITY

35 In the moist air system of humidifying it is evidently essential, as shown in Table 1, that the difference between the dewpoint temperature of the incoming air and the room temperature

shall not exceed a predetermined value depending upon the percentage of humidity to be maintained. The minimum temperature at which air can be introduced is evidently the dewpoint or saturation temperature at the apparatus. This permissible temperature rise limits the possible cooling effect to be obtained from each cubic foot of air as shown in Table 5. This relationship is of primary importance in the design of the humidifying system and the disregard of it has been the chief cause of failure or of unsatisfactory operation.

36 In the majority of industrial applications the problem during warm weather, and in some instances throughout the entire year, is as much a question of cooling as of humidifying. Indeed, in the

TABLE 5 COOLING CAPACITY OF CARRIER HUMIDIFYING SYSTEM

Per Cent Humidity in Room	Difference between Dewpoint and Room Temperature	Cu. Ft. of Air at 70 Deg. Fahr. Required per B.t.u. Cooling Effect
50	20.3	2.71
55	17.7	3.11
60	15.2	3.63
65	12.8	4.31
70	10.8	5.10
75	8.8	6.27
80	6.8	8.11

moist air system, as has just been shown, one is dependent on the other. In every industrial air-conditioning plant there are four sources of heat which must be taken into account in the design of the system:

a Radiation from the outside owing to the maintenance of a lower temperature inside. At ordinary humidities this is negligible, but at high humidities and in dehumidifying plants it is an important factor, owing to the increased temperature difference. This may be calculated from the usual constants of radiation.

b The heating effect of direct sunlight. This is especially noticeable from window shades and exposed windows and skylights where the entire heat energy of the sunlight is admitted to the room, and from the roof which constitutes the greater amount of sunlight exposure and which in the ordinary construction transmits heat much more readily than the walls. Precautions should be taken

where high humidities are desired to shade exposed windows and to insulate the roof thoroughly. Ventilators in the roof are of great advantage in removing the hot layer of air next it and those of ample capacity should always be provided.

c The radiation of heat from the bodies of the operatives. This amounts to about 400 to 500 B.t.u. per operative, about one-half of which is sensible heat, the other half being transformed into latent heat through evaporation.

d The heat developed by power consumed in driving the machinery and in the manufacturing processes in general. According to the laws of conservation of energy, all power used in manufacturing is ultimately converted entirely into its heat equivalent. Each horsepower of energy therefore creates 42½ B.t.u. of heat per minute which must be cared for by ventilation. In high-powered mills this is the chief source of heating and is frequently sufficient to overheat the building even in zero weather, thus requiring cooling by ventilation the year round.

TABLE 6 POSSIBLE ROOM TEMPERATURES OBTAINABLE IN CARRIER SYSTEM OF HUMIDIFYING, AT VARIOUS OUTSIDE WET-BULB TEMPERATURES AND VARIOUS PERCENTAGES OF HUMIDITY IN THE ROOM

Outside Wet-Bulb Temperature	Percentage Humidity in Room					
	55	60	65	70	75	80
50	67.0	64.3	62.0	59.8	57.7	56.0
55	72.0	69.6	67.5	65.2	63.1	61.1
60	77.5	74.9	72.5	70.4	68.3	66.3
65	82.8	80.2	77.6	75.6	73.5	71.6
70	87.2	85.6	83.1	80.0	78.7	76.7
75	93.7	90.8	87.3	85.9	83.9	81.9
80	99.1	96.2	93.4	91.2	89.0	87.0

RELATION OF ROOM TEMPERATURE TO OUTSIDE WET-BULB TEMPERATURE

37 During cool weather the dewpoint or saturation temperature at the apparatus is secured and controlled artificially at whatever point required. During warm weather, however, it is impossible during the greater part of the time to obtain as low a dewpoint as desired without refrigeration, which in the majority of cases of humid-

ifying is impracticable. The lowest saturation temperature that can be obtained is the same as the outside wet-bulb temperature, as has been shown. Therefore, the dewpoint in the room will always be the same as the outside wet-bulb temperature. Since the difference between the dewpoint and the room temperature is dependent upon the percentage of relative humidity maintained, the minimum room temperature and the percentage of humidity required in the enclosure will be as shown in Table 6. It will be noted that the higher the humidity carried, the lower the room temperature may be kept.

TABLE 7 B.T.U. REFRIGERATION REQUIRED TO COOL 1000 CU. FT. OF AIR (MEASURED AT 70 DEG.) FROM A GIVEN WET-BULB TEMPERATURE TO A GIVEN DEWPOINT

Leaving Dewpoint	Ammonia Temperature	Suction Pressure (Gage)	Per Cent Compressor Ratings at 15 Lb.	Per Cent Horsepower Compared with Horsepower Required at Rated Suction Pressure of 15 Lb. Gage	Entering Wet-Bulb Temperature							
					50	55	60	65	70	75	80	85
65	45	65.96	270	41.5	.....	.....	.....	.....	296	606	961	1350
60	40	58.29	244	45.5	.....	.....	.....	259.0	553	865	1220	1609
55	35	51.22	220	49.5	.....	.....	221.5	480.5	777	1086	1440	1840
50	30	44.72	199	54.5	.....	203	425.0	683.0	980	1290	1570	2030
45	25	38.73	182	59.5	185	388	611.0	869.0	1165	1474	1830	2220
40	20	33.25	164	66.0	359	569	791.0	1050.0	1345	1656	2010	2400

ELEMENTS OF DESIGN OF SPRAY TYPE OF DEHUMIDIFIERS

38 Dehumidifiers may be of the spray type previously described or of the surface type. A knowledge of the relation of water temperature to the leaving air temperature in either type is essential. In the spray type of one stage having two banks of opposed nozzles, the air temperature leaving is practically identical with the temperature of the leaving water, the difference never exceeding 1 deg. in a properly designed apparatus. The air will always be saturated when leaving and under some conditions there is a slight tendency to entrainment even after thorough elimination.

39 The degree of entrainment is dependent upon the range of temperature of both the air and the water. In general the smaller the temperature range, the less the tendency is to moisture entrainment or supersaturation. This may be reduced where a

considerable lowering of air temperature is required by passing it successively through two or more dehumidifiers in series. When the system is properly designed the entrainment should not be sufficient to raise the true dewpoint temperature more than 1 deg.

REFRIGERATION REQUIRED FOR DEHUMIDIFYING

40 The heat to be removed in cooling a known weight of air from a given temperature and moisture content to a given dewpoint temperature is evidently the difference of the total heat quantities contained in the air under these respective conditions. These values of total heat are given in Figs. 1 and 2 of the accompanying paper on Rational Psychrometric Formulae.[1] It is there shown that the total of latent and specific heat in 1 lb. of pure air is dependent upon the wet-bulb temperature only. Table 7 shows the amount of refrigeration required to cool and dehumidify 1000 cu. ft. of air between various given wet-bulb temperatures and final dewpoints.

41 The amount of water required to cool air in a one-stage spray system dehumidifier may be calculated from the foregoing data as follows:

$$W(t_W - t_2) = N(H_1 - H_2) = N(t'_1 - t_2)\frac{\Delta H}{\Delta t'} \quad \ldots\ldots\ldots [3]$$

$$W = N\left(\frac{H_1 - H_2}{t_W - t_2}\right) = N\left(\frac{t'_1 - t_2}{t_W - t_2}\right)\frac{\Delta H}{\Delta t'} \quad \ldots\ldots\ldots [4]$$

where

W = weight of water in lb.
N = weight of air in lb.
t_W = initial water temperature
t'_1 = initial wet-bulb temperature of air
t_2 = final dewpoint temperature of air
H_1 = initial total heat in 1 lb. of air at wet-bulb temperature, t'
H_2 = total heat in 1 lb. of air at final dewpoint t_2
$\frac{\Delta H}{\Delta t'}$ = approximate rate of total heat change at the given temperature per degree change in wet-bulb temperature, t'

42 To find the final temperature possible with a given weight of water and of air at temperature t_W and t'_1, respectively, we have from [3]

[1] The Journal, November 1911, p. 1309.

$$Wt_W - Wt_2 = Nt'_1 \frac{\Delta H}{\Delta t'} - Nt_2 \frac{\Delta H}{\Delta t'} \quad [5]$$

Hence

$$t_2 = \frac{Nt'_1 \frac{\Delta H}{\Delta t'} - Wt_W}{N \frac{\Delta t}{\Delta t'} - W} \quad [6]$$

ECONOMY OF THE SPRAY TYPE OF DEHUMIDIFIER

43 The chief advantage, aside from economy of space, of the spray type of dehumidifier over the surface type, lies in its ability to bring the spray water and the air to substantially the same temperature, while in the surface type there is usually from 15 to 25 deg. difference. This permits a much increased efficiency of cooling with the spray type where artesian well water is used. When the spray water is cooled by artificial refrigeration, using the compression system, this higher water temperature permits a much increased ammonia pressure in the water cooler, often doubling the absolute pressure. As shown in Table 7, this increases the capacity of the ammonia compressor correspondingly and greatly reduces the horsepower required per ton of refrigeration. The ammonia condenser, of course, remains unchanged.

HEAT INTERCHANGER

44 When the reduction of the moisture contents is of more importance than the cooling effect, a heat interchanger or economizer may be used to great advantage. In this apparatus the cold air leaving the dehumidifier is passed through the inside of a system of tubes while the warm incoming air is passed over the outside of these tubes and in the opposite direction. This permits an effectual transfer of heat, often cooling the incoming air to a lower temperature than that of the dehumidified air leaving the interchanger. The saving effected in this way is often more than one-third of the total refrigeration. The calculation of the dehumidifying surface required may be effected by the use of the formula for heat transmission given in Par. 91. Precaution must be taken in this calculation, however, to note that the convection resistances on each side of the surface must be taken into account, as the rate of heat transmission is greatly reduced on this account.

AIR COOLING AND DEHUMIDIFYING WITH COOLING COILS

45 Calculation of the cooling effect of condensing coils for different velocities through the clear area and various temperature differences is fully discussed under the theory of convection in Par. 62 to 75. The same formulae may be applied in both cases.

46 Condensing coils give a much more rapid rate of conductivity per degree difference in temperature than steam coils, owing to the additional effect of condensation. This relationship however has

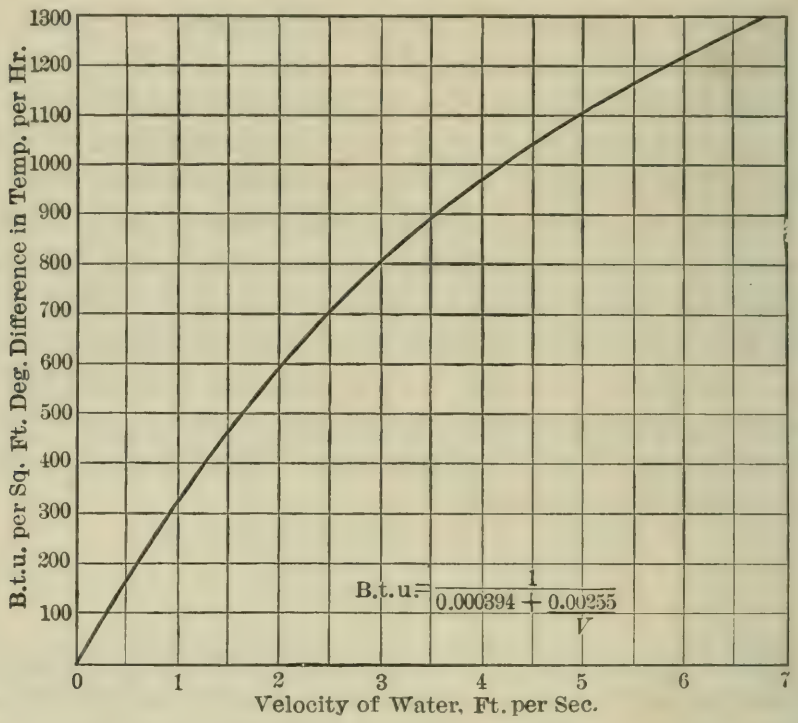


FIG. 28 HEAT TRANSMISSION FROM STEAM TO WATER

never before been investigated nor have there been any data previously available. The fact that the moisture contents of the air may be greatly reduced without leaving it saturated is explained and means for the accurate calculation of the moisture contents are given.

47 Where the air is simply cooled and no condensation takes place, the rate of heat transmission from cooling water to air is the exact inverse of the sum of the convection resistances of both the air and the water films.

48 We have the rate of heat conduction between steam and water approximately given by the curve shown in Fig. 28, as plotted by the

authors from condenser tests. This gives for the rate of conductivity for steam to water

$$K = \frac{1}{0.000394 + \frac{0.00255}{V}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [7]$$

where V = velocity in ft. per sec.

49 When V_W = velocity of the water in feet per minute, the resistance through a conducting wall from steam to water is

$$R_\Sigma = \frac{1}{K_W} = 0.000394 + \frac{0.153}{V_W} \quad \ldots\ldots\ldots\ldots [8]$$

Of the constant factor, 0.000394, the resistance between the surface films, probably considerably less than one-half is due to the water film alone, or less than 0.0002. For heat transmission from steam to air we have, as shown later

$$K_A = \frac{1}{0.0447 \frac{\theta'_m}{625} + \frac{50.66}{V_o}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [9]$$

where

θ'_m = mean absolute temperature of air film

V_o = velocity through clear area (measured at 70 deg.)

50 Therefore we have for the convection resistance from steam to air

$$R_\Sigma = \frac{1}{K_A} = 0.0447 \frac{\theta'_m}{625} + \frac{50.66}{V_o} \quad \ldots\ldots\ldots\ldots [10]$$

51 If we take 60 deg. fahr. or 520 deg. absolute as a fair average mean temperature of the air film, we will have

$$R_\Sigma = \frac{1}{K_A} = 0.0373 + \frac{50.66}{V_o} \quad \ldots\ldots\ldots\ldots [11]$$

52 It is noticeable that the combined resistance of the steam and air films is 100 times as great as the resistance of the combined steam and water films, and hence very probably more than 200 times as great as the water film alone.

53 We may take as the entire convection resistance between air and water, when there is no condensation

$$R_{\Sigma} = \frac{1}{K_W} + \frac{1}{K_A} = 0.0002 + 0.0373 + \frac{0.153}{V_W} + \frac{50.66}{V_o} \quad \ldots\ldots [12]$$

$$R_{\Sigma} = \frac{1}{K_{\Sigma}} = 0.0375 + \frac{0.153}{V_W} + \frac{50.66}{V_o} \quad \ldots\ldots\ldots [13]$$

where

V_W = velocity of water through cooling pipes in ft. per min.

V_0 = velocity of air through minimum clear area between pipes in ft. per min.

54 The heat transmission then between air and water when no condensation occurs for any initial and final temperatures will then be, in B.t.u. per hour

$$H = K_{\Sigma} D_m$$

or

$$H = \frac{(\theta_{A_1} - \theta_{W_1}) - (\theta_{A_2} - \theta_{W_2})}{\left(0.0375 + \frac{0.153}{V_W} + \frac{50.66}{V_o}\right) \log_e \left(\frac{\theta_{A_1} - \theta_{W_1}}{\theta_{A_2} - \theta_{W_2}}\right)} \quad \ldots\ldots [14]$$

When D_m is the mean temperature difference between steam and air, $(\theta_{A_1} — \theta_{W_1})$ is the initial temperature difference between air and water. The final temperature difference between air and water is $(\theta_{A_2} — \theta_{W_2})$. (See Appendix No. 2.)

RATE OF TRANSMISSION BETWEEN AIR AND WATER WHERE CONDENSATION OCCURS

55 Whenever the relation between the water temperature and the temperature of the air is such that the film temperature is below the dewpoint, then condensation will occur. The air in the surface film will always be saturated at the temperature of the surface film, while the main body of the air passing through the heater may be considerably above the saturation point, even after considerable moisture has been condensed out from it, the final state of the air being simply the result of a mixture of air saturated at the mean film temperature with the main body of the air. For proof of this, see the discussion of the laws of convection in Pars. 62 to 75.

56 Actual tests with condensing coils, as well as the theory, both indicate that the resistance of the surface film of the air and the temperature of the surface film are in no way affected by the condensation of moisture. However, the rate of convection from the water to the conducting wall is proportionately affected by the increased

amount of heat transmitted, on account of condensation. As the effect of the rate of convection from water to the conducting wall is relatively slight, it may be neglected in approximate calculations, and the cooling effect upon the air will be approximately the same as though no moisture were condensed. The rate of heat transmission in a surface dehumidifier will be increased in proportion to the amount of heat given up by condensation. Proper allowance must of course be made for the increased rise in water temperature.

57 The total rate of heat transfer may be calculated more accurately by taking into account the increased rate of transmission from the conducting wall to the air as follows:

$$H = \frac{(1+m)\,(\theta_A - \theta_W)\ \text{mean}}{(1+m)\,R_{\Sigma W} + R_{\Sigma A}} \quad \text{(see Appendix No. 4)} \quad \ldots\ldots\ldots\ldots [15]$$

$$H = \frac{(1+m)\,(\theta_A - \theta_W)\ \text{mean}}{\left[(1+m)\,\frac{0.153}{V_W}\right] + \left[0.0375 + \frac{50.66}{V_o}\right]} \quad \ldots\ldots\ldots\ldots\ldots\ldots [16]$$

$$H = \frac{(1+m)\,[(\theta_{A_1} - \theta_{W_1}) - (\theta_{A_2} - \theta_{W_2})]}{\left[0.0375 + \frac{(1+m)0.153}{V_W} + \frac{50.66}{V_o}\right] \log_e \left(\frac{\theta_{A_1} - \theta_{W_1}}{\theta_{A_2} - \theta_{W_2}}\right)} \quad \ldots\ldots\ldots [17]$$

where

$$m = \frac{\Delta\ (\text{latent heat per lb. of air})}{\Delta\ (\text{sensible heat per lb. of air})}$$

MOISTURE CONTENT OF AIR LEAVING SURFACE DEHUMIDIFIER

58 As we have previously shown, air leaving the surface dehumidifier need not necessarily be saturated, even though the moisture content be considerably reduced. Let θ_{A_1} be the initial air temperature and θ_{A_2} be the final air temperature. Let W_1 be the initial weight of water in a pound of pure air, and W_2 be the final weight of water contained in 1 lb. of pure air. Let θ'_m be the mean temperature of the surface film, for calculation of which see Appendix No. 2. Let W'_m be the weight of water contained in one pound of pure air saturated at the mean film temperature. Then we will have

$$\theta_2 = (1 - x)\,\theta_1 + x\theta'_m \quad \ldots\ldots\ldots\ldots\ldots [18]$$

$$W_2 = (1 - x)\,W_1 + xW'_m \quad \ldots\ldots\ldots\ldots\ldots [19]$$

Hence, eliminating x, we will have

$$\frac{W_1 - W_2}{\theta_1 - \theta_2} = \frac{W_1 - W'_m}{\theta_1 - \theta'_m} \quad \ldots\ldots\ldots\ldots\ldots\ldots [20]$$

$$W_2 = W_1 - \frac{(W_1 - W'_m)\ (\theta_1 - \theta_2)}{\theta_1 - \theta'_m} \quad \ldots\ldots\ldots\ldots [21]$$

RATE OF HEAT TRANSMISSION IN INDIRECT SURFACE AIR HEATERS AND DEHUMIDIFIERS

59 An essential element in air-conditioning apparatus is the indirect radiation necessary to warm the air after saturation in the humidifier. In systems where the air is required for ventilation only, and the heating is effected by separate radiation, as in many of the larger installations, it is necessary to use indirect radiation only for the purpose of warming the air from the dewpoint temperature to approximately room temperature. In the majority of installations, however, the air is heated by means of indirect radiation to the point required to maintain the proper room temperature.

60 The temperature to which this conditioned air must be heated evidently depends on the outdoor temperature and upon the temperature difference to be maintained between the room and the dewpoint of the air supply, as required by the humidity control. On this account data pertaining to the proper design of such an indirect heating surface are of especial importance. These data are also applied in air conditioning to the design of surface coolers, dehumidifiers, and interchangers.

61 The factors of importance in such a design are the relation of free area to heating surface; the rate of heat transmission for various velocities through the clear area, and various temperature differences between the air and heating medium; and the resistance of the heater to the passage of air. The following experimental and mathematical investigation was conducted for the purpose of establishing these relationships, as very little authoritative information is at present available on this subject.

THEORY OF CONVECTION WITH FORCED CIRCULATION

62 The accompanying diagram, Fig. 29, presents a graphical representation of the process of the heat transfer from steam to air through conducting wall. Experimental investigation leads us to

conclude that the exterior of the conducting wall is covered with a surface film into which heat passes directly from the conducting wall, and whose resistance to the passage of heat is independent of the velocity of the convecting medium; but it is a direct function of the density and specific heat of that medium.

63 The total resistance of the surface film of the steam, of the conducting wall, and of the surface film of the conducting medium, may therefore be represented by a constant R, which is independent of the temperature difference between the steam and the convecting medium, and of the velocity of the latter.

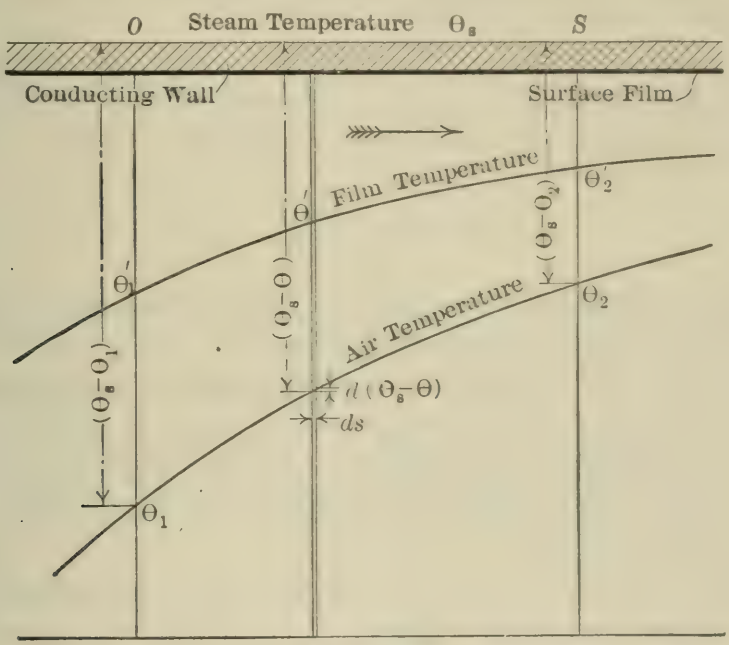


FIG. 29 IDEAL DIAGRAM ILLUSTRATING THE THEORY OF HEAT TRANSMISSION

64 Experimental investigation also indicates that heat is transferred from the surface film to the main body of the convecting medium, that is to the air, entirely by displacement. Particles of air in the surface film are displaced by impact, due to the velocity of the air over the surface, and are thus mixed with the main body of the air. This displacement may be shown to be in direct proportion to the velocity. The rate of heat transfer from the surface film to the air is therefore directly proportional to the product of the velocity and temperature difference between the film and air.

65 Let θ_s represent the steam temperature, the distance OS the extent of surface over which a unit of air passes progres-

sively. Let θ_1 θ_2 represent the progressive air temperatures with reference to the steam temperature θ_s, and the line θ'_1 θ'_2 the corresponding surface film temperatures. Then $\theta_s - \theta$ represents the difference between air temperature and steam temperature at any instant, and $\theta s - \theta'$, the corresponding difference between the steam temperature and the film temperature.

66 The foregoing theory may then be stated mathematically as

$$\frac{dH}{dt} = \frac{1}{R}(\theta_s - \theta') \quad \ldots\ldots\ldots\ldots\ldots\ldots [22]$$

where $\frac{dH}{dt}$ is the rate of transfer of heat per unit of surface through the resistance R of the separating wall and surface films of steam and air. Assuming that this rate is proportional to the difference in temperature between the two films, from [22] we have

$$\theta' = \theta_s - R\frac{dH}{dt} \quad \ldots\ldots\ldots\ldots\ldots\ldots [23]$$

67 According to both theoretical considerations and experimental evidence R probably varies inversely with the density of the air film. Hence with increased temperature of the surface film we would expect R to be increased. This variation, however, for approximate calculations is negligible, although in exact calculations a correction must be made. This correction becomes important when the temperature range is great.

68 Assuming the number of particles from the surface film to be directly proportional to the velocity, the rate of heat convection from the surface film is

$$\frac{dH}{dt} = \frac{C_p W V(\theta' - \theta)}{B} \quad \ldots\ldots\ldots\ldots\ldots\ldots [24]$$

where C_p is the specific heat of the air in the surface film, W is the density of the air at the temperature θ_A, V the velocity in feet per minute through the clear area, and B a constant to be determined experimentally, dependent upon the path of the air and upon the form and arrangement of the convecting surface.

69 From [24] we have

$$\theta' = \frac{B}{C_p W V}\frac{dH}{dt} + \theta \quad \ldots\ldots\ldots\ldots\ldots\ldots [25]$$

Equating [23] and [25] we have

$$\left(\frac{B}{C_p WV}+R\right)\frac{dH}{dt}=\theta_s-\theta \quad \ldots\ldots\ldots\ldots\ldots [26]$$

$$\frac{dH}{dt}=\frac{\theta_s-\theta}{R+\dfrac{B}{C_p WV}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [27]$$

70 Let W_0 and V_0 be the corresponding densities and velocities of the air at an absolute base temperature θ_0 and let θ be the absolute temperature of the air corresponding to W and V then

$$WV=\left(\frac{W_0\theta_0}{\theta}\right)\left(\frac{V_0\theta}{\theta_0}\right)=W_0V_0$$

Hence, substituting in [27] we have

$$\frac{dH}{dt}=\frac{\theta_s-\theta}{R+\dfrac{B}{C_pW_0V_0}} \quad \ldots\ldots\ldots\ldots\ldots [28]$$

which shows that the term $\dfrac{B}{C_pW_0V^0}$ is unaffected by variation in temperature; and assuming R constant, that $\dfrac{dH}{dt}$ is directly proportional to the difference in temperature between steam and air.

71 Therefore, let K be the rate of transmission in B.t.u. per sq. ft. per hour per degree difference in temperature between the steam and air. Then

$$K=\frac{1}{R+\dfrac{B}{C_pW_0V_0}} \quad \ldots\ldots\ldots\ldots\ldots [29]$$

and

$$\frac{dH}{dt}=K(\theta_s-\theta) \quad \ldots\ldots\ldots\ldots\ldots [30]$$

72 Let H_s be the total heat transferred per hour from a surface S, then

$$H_s=K(\theta_s-\theta)S \quad \ldots\ldots\ldots\ldots\ldots [31]$$

$$dH_s=K(\theta_s-\theta)dS \quad \ldots\ldots\ldots\ldots\ldots [32]$$

Also

$$dH_s=-C_pGd(\theta_s-\theta) \quad \ldots\ldots\ldots\ldots\ldots [33]$$

where

G = weight of air per hour passing over the surface S.

73 Equating [32] and [33] we have

$$K(\theta_s - \theta)\,dS = -C_p G d(\theta_s - \theta) \quad \ldots\ldots\ldots\ldots [34]$$

Integrating between limits $(\theta_s - \theta_1)$ and $(\theta_s - \theta_2)$ and between O and S, we have

$$\int_o^S K dS = \int_{(\theta_s-\theta_1)}^{(\theta_s-\theta_2)} -\frac{C_p G d(\theta_s - \theta)}{(\theta_s - \theta)} \quad \ldots\ldots\ldots\ldots [35]$$

$$KS = C_p G \log_e \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) \quad \ldots\ldots\ldots\ldots [36]$$

$$G = 60\ AWV = 60\ AW_oV_o$$

where

A = clear area through heater having surface S
V = velocity in ft. per min. through clear area
W = density of air in lb. per cu. ft.

Hence

$$\log_e \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) = \frac{KS}{60 C_p A W_o V_o} \quad \ldots\ldots\ldots\ldots [37]$$

Changing this to common logarithms

$$\log_{10} \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) = \frac{KS}{(2.3026 \times 60) C_p A W_o V_o} \quad \ldots\ldots\ldots [38]$$

74 From [29] we have

$$K = \frac{1}{R + \dfrac{B}{C_p W_o V_o}}$$

$$\log_{10} \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) = \frac{S}{(2.3026 \times 60) A C_p W_o \left(R V_o + \dfrac{B}{C_p W_o}\right)} \quad \ldots [39]$$

$$= \frac{S}{(2.3026 \times 60)(R C_p W_o Q + BA)} \quad \ldots\ldots\ldots [40]$$

where Q = cu. ft. of air per minute at standard temperature. This will also reduce to

$$\log_{10} \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) = \frac{f}{mV + n} \quad \ldots\ldots\ldots\ldots [41]$$

where

$f=\frac{S}{A}$, n is a constant, and m is substantially a constant except as varied by change in the absolute temperature of the surface film.

SURFACE FILM TEMPERATURE

75 From equation [23] we have for the surface film temperature

$$\theta' = \theta_s - R\frac{dH}{dt}$$

From equation [28]

$$\frac{dH}{dt} = \frac{\theta_s - \theta}{R + \frac{B}{C_p W_o V_o}}$$

Substituting this value in equation [23] we have

$$\theta' = \theta_s - R\left(\frac{\theta_s - \theta}{R + \frac{B}{C_p W_o V_o}}\right) \quad \ldots\ldots\ldots\ldots\ldots [42]$$

Hence

$$\theta' = \theta_s - RK(\theta_s - \theta) \quad \ldots\ldots\ldots\ldots\ldots [43]$$

and

$$R = \frac{1}{K}\left(\frac{\theta_s - \theta'}{\theta_s - \theta}\right) \quad \ldots\ldots\ldots\ldots\ldots [44]$$

Also

$$\theta_s - \theta' = RK(\theta_s - \theta) \quad \ldots\ldots\ldots\ldots\ldots [45]$$

i.e., the difference between the steam and film temperature is proportionate to the difference between the steam and air temperatures when RK is constant, hence approximately so when V_0 is constant.

EXPERIMENTAL DETERMINATION OF THE LAWS OF CONVECTION

76 We will now proceed to determine the constants in the foregoing equations and give the experimental evidence by which the above theory may be corroborated. Table 8 gives the observed and corrected results from ten heat transmission tests conducted upon

TABLE 8 LOG OF OBSERVED AND CORRECTED RESULTS FROM HEAT TRANSMISSION TESTS

Test No.	1			2			3			4			5		
	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis
Revolutions Per Minute	80	.....	.....	120	.....	.....	160	.....	.....	200	.....	.....	240	.....	.....
Barometer, In	29.46	.....	.....	29.40	.....	.....	29.45	.....	.....	29.47	.....	.....	29.43	.....	.....
Velocity through Clear Area, Ft. per Min	335	276	254	453	375	345	594	499	459	738	626	576	950	805	741
Initial Temperature of Air	16.5	20.0	.....	20.0	20.0	.....	19.0	20.0	.....	16.5	20.0	.....	21.0	20.0	.....
Final Temperature of Air	174.4	183.4	.....	170.4	176.4	.....	162.5	167.6	.....	156.5	160.8	.....	155.0	157.1	.....
Temperature Rise	157.9	163.4	.....	150.4	156.4	.....	143.5	147.6	.....	140.0	140.8	.....	134.0	137.1	.....
Total Air per Min. Cu. Ft.	3440	2830	2604	4650	3850	3542	6100	5110	4702	7570	6420	5906	9750	8260	7600
Total Air per Min. Lb.	212	212	195	288	288	265	383	383	352	481	481	443	619	619	570
Condensation per Section, Lb. per ½-Hr. Run: 1st	58.5	59.2	.....	66.4	68.3	.....	82.2	83.8	.....	97.0	97.8	.....	104.7	108.2	.....
2d	45.3	46.2	.....	55.0	56.8	.....	69.7	71.5	.....	83.5	84.5	.....	91.4	94.3	.....
3d	34.6	35.4	.....	43.4	45.0	.....	57.5	59.0	.....	69.2	70.1	.....	76.7	79.2	.....
4th	28.6	29.4	.....	36.0	37.4	.....	49.1	50.5	.....	60.1	61.0	.....	66.3	68.6	.....
5th	24.2	25.0	.....	30.9	32.2	.....	42.9	44.1	.....	53.1	54.0	.....	60.6	62.9	.....
6th	17.3	18.0	.....	23.3	24.5	.....	33.8	35.0	.....	42.4	43.2	.....	49.9	51.9	.....
7th	14.8	15.6	.....	19.6	20.7	.....	28.5	29.6	.....	36.1	36.9	.....	43.2	45.0	.....
8th	12.2	12.7	.....	16.4	17.3	.....	24.1	25.0	.....	30.8	31.4	.....	37.6	39.1	.....
Total Condensation ½ Hr.	.....	241.5	.....	.....	302.2	.....	.....	398.5	.....	.....	478.9	.....	.....	549.2	.....
Total Condensation 1 Hr.	.....	483.0	.....	.....	604.4	.....	.....	797.0	.....	.....	957.8	.....	.....	1098.4	.....
B.t.u. Delivered per Lb. of Condensation	.....	977.0	.....	.....	979.4	.....	.....	982.3*	.....	.....	979.0	.....	.....	976 1	.....
B.t.u. per Sq. Ft. Heater per Hr. from the Steam	.....	466.0	.....	.....	583.0	.....	.....	773.0	.....	.....	926.0	.....	.....	1058.0	.....
B.t.u. per Sq. Ft. per Hr. absorbed by the Air	.....	496.0	456	.....	644.0	592	.....	808.0	743	.....	967.0	890	.....	1214.0	1117.0

TABLE 8—Continued

Test No.	6			7			8			9			10		
	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per Cent Correction Basis	Test Conditions	Reduced to Standard Conditions	Reduced to 8 Per cent Correction Basis
Revolutions Per Minute	280	.....	.....	320	.....	.....	360	.....	.....	360	.....	.....	400	.....	.....
Barometer, In.	29.40	.....	.....	29.43	.....	.....	29.43	.....	.....	29.43	.....	.....	29.40	.....	.....
Velocity through Clear Area, Ft. per Min.	1067	911	838	1199	1027	945	1315	1135	1044	1345	1165	1072	1491	1286.0	1183
Initial Temperature of Air	20.5	20.0	.....	21.5	20.0	.....	20.6	20.0	.....	19.5	20.0	.....	19.8	20.0	.....
Final Temperature of Air	150.1	152.0	.....	148.6	149.8	.....	142.6	143.5	.....	143.6	145.4	.....	142.6	143.9	.....
Temperature Rise	129.6	132.0	.....	127.1	129.8	.....	122.0	123.5	.....	124.1	125.4	.....	122.8	123.9	.....
Total Air per Min. Cu. Ft.	10950	9350	8602	12300	10520	9679	13500	11650	10720	13800	11950	10995	15300	13200.0	12145
Total Air per Min. Lb.	701	701	645	789	789	726	873	873	803	895	895	823	989	989	910
Condensation per Section, Lb. per ½-Hr. Run 1st	112.2	115.0	.....	130.6	134.5	.....	140.2	143.1	.....	136.2	139.1	.....	145.7	149.2	.....
2d	107.0	110.0	.....	114.1	117.8	.....	113.0	116.0	.....	123.6	126.7	.....	135.1	138.6	.....
3d	87.4	90.0	.....	97.6	101.0	.....	109.6	112.8	.....	107.1	109.8	.....	115.7	118.8	.....
4th	76.9	79.3	.....	87.6	90.6	.....	99.1	102.0	.....	96.2	98.6	.....	103.8	106.4	.....
5th	68.6	70.9	.....	78.1	80.8	.....	89.5	92.2	.....	86.5	88.7	.....	89.9	92.5	.....
6th	57.4	59.4	.....	64.4	66.7	.....	74.1	76.4	.....	72.7	74.7	.....	80.1	82.5	.....
7th	50.2	50.0	.....	57.5	59.7	.....	66.9	69.0	.....	66.1	68.0	.....	73.4	75.6	.....
8th	43.6	45.2	.....	50.7	52.5	.....	59.9	61.8	.....	58.6	60.2	.....	64.1	66.0	.....
Total Condensation ½ Hr.	.....	621.8	.....	.....	703.6	.....	.....	773.3	.....	.....	765.8	.....	.....	829.6	.....
Total Condensation 1 Hr.	.....	1243.6	.....	.....	1407.2	.....	.....	1546.6	.....	.....	1531.6	.....	.....	1659.2	.....
B.t.u. Delivered per Lb. of Condensation	.....	977.1	.....	.....	968.9	.....	.....	972.2	.....	.....	970.3	.....	.....	975.1	.....
B.t.u. per Sq. Ft. Heater per Hr. from the Steam	.....	1200.0	.....	.....	1345.0	.....	.....	1483.0	.....	.....	1470.0	.....	.....	1595.0	.....
B.t.u. per Sq. Ft. per Hr. absorbed by the Air	.....	1322.0	1216	.....	1466.0	1349	.....	1546.0	1422	.....	1606	1477	.....	1753.0	1613

eight sections of fan system heaters, each section being composed of four rows of 1-in. wrought-iron pipe, making a total depth of 32 rows of pipe. Details of the heater construction are shown in Fig. 30. For complete description of tests and corrections applied see Appendix No. 1.

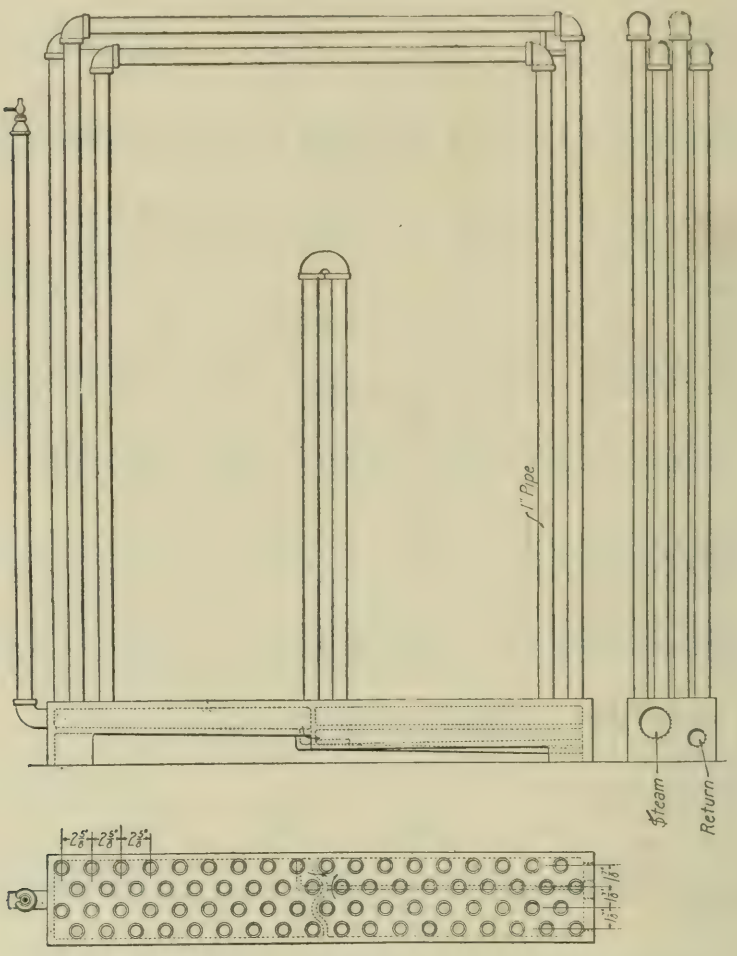


FIG. 30 DETAIL OF HEATER CONSTRUCTION

77 Curves shown in Fig. 31 are plotted from the values given in column 2 of each test in Table 8, which give air velocities through the clear area based on actual air measurements corrected to corresponding volume at 70 deg. fahr. and 29.92 in. barometric pressure. The lower curve gives the rate of heat transmission as determined from the condensation. The upper curve gives the corresponding values as determined from observed air measurements. These two curves show a uniform discrepancy of approximately 8 per cent, as indicated in Fig. 32. As both the temperature and condensation measurements

were made with considerable care, and since the per cent of error is substantially uniform at all velocities, it is evident that a systematic error was made of 8 per cent in the measurement of the air volume. This may be accounted for in large measure by the fact that no allowance was made for the coefficient of the discharge orifice. Accord-

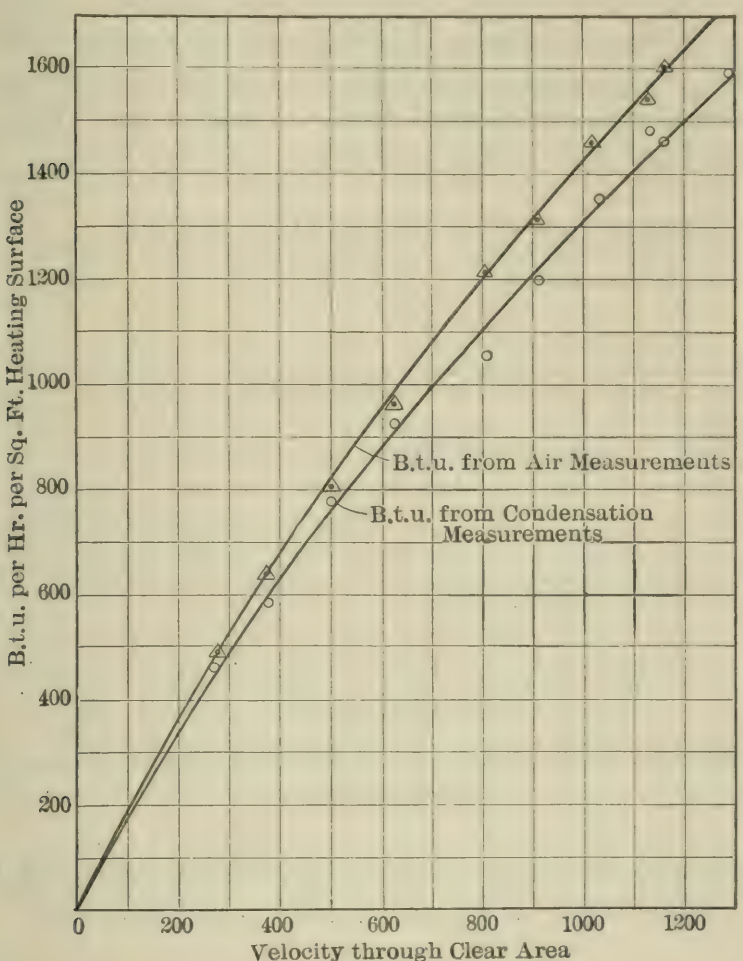


FIG. 31 EFFECT OF VELOCITY UPON RATE OF TRANSMISSION FROM ORIGINAL AIR MEASUREMENTS

ingly both the rate of transmission from the air measurements and the corresponding velocities have each been reduced 8 per cent.

78 From these corrected values and from the condensation, the combined curve shown in Fig. 33 has been plotted, showing a remarkably close agreement of both values, thus thoroughly establishing the true form of the curve.

COEFFICIENT OF TRANSMISSION

79 From the above data we may calculate K, that is the rate of transmission per square foot per degree difference in temperature at the various velocities, providing we may assume, as in the foregoing theory, that the rate of transmission at any instant is proportionate to the temperature difference between the steam and air. To establish this relationship Table 9, containing the corrected test data on the individual sections, is given, and also Fig. 34 in which the transmission curves are plotted from the values in Table 9. These curves have been derived through the well-known systems of fairing,

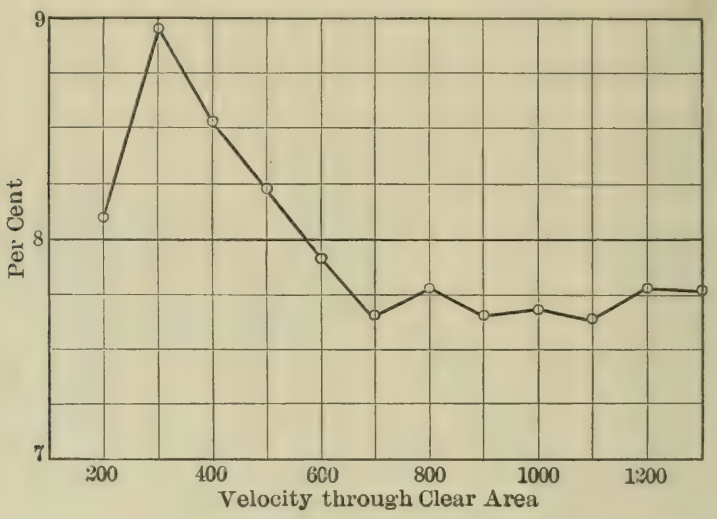


FIG. 32 PERCENTAGE OF ERROR IN AIR MEASUREMENTS AS COMPARED WITH CONDENSATION

and of cross and increment plotting, until they represent the most probable relationships to be determined experimentally.

80 From Fig. 34 the B.t.u. values given in the second column of Table 10 have been taken. Values exhibited here are taken at a velocity of 1000 ft. per min. through the clear area. While comparisons are given here for only one velocity, it should be understood that the same comparisons have been made at other velocities. Column 3 gives the temperature rise in each consecutive section, as computed from the experimental rate of transmission given in column 2, this relationship being expressed by the formula

$$\theta_2 - \theta_1 = \frac{H'S}{60\,C_p A W_o V_o}$$

where H' = B.t.u. per hr. per sq. ft. The fourth column gives the

computed temperature leaving the consecutive sections. From column 4, columns 5 and 6 are computed. Column 7 gives the ratio of the initial temperature difference between steam and air to the final temperature difference between the steam and air, for each consecutive section. From this comparison it will be seen that this ratio $\frac{\theta_s - \theta_1}{\theta_s - \theta_2}$ is substantially constant in all sections, for any one velocity, regardless of the variation in the entering temperature. If the rate of transmission per degree difference in temperature between steam and air was absolutely constant at any one velocity, it could be shown from equation [38] that the ratio of $\frac{\theta_s - \theta_1}{\theta_s - \theta_2}$ for any section as well as $\log \frac{\theta_s - \theta_1}{\theta_s - \theta_2}$ would also be absolutely constant, and conversely.

TABLE 9 B.T.U. PER SQ. FT. OF SURFACE PER HOUR, CALCULATED FROM CONDENSATION

Section Number	Revolutions Per Minute									
	80	120	160	200	240	280	320	360	360	400
	Observed Velocities Corrected to Standard Conditions and Reduced 8 Per cent									
	254	345	459	576	741	838	945	1044	1072	1183
1	912	1060	1278	1518	1670	1780	2029	2205	2130	2300
2	713	880	1110	1311	1458	1705	1800	1785	1938	2135
3	546	697	915	1089	1224	1397	1545	1735	1680	1830
4	453	579	783	944	1060	1229	1387	1570	1510	1640
5	386	499	685	836	971	1099	1238	1420	1358	1425
6	268	379	544	669	801	918	1020	1178	1143	1271
7	241	321	460	571	695	805	911	1063	1040	1165
8	196	267	388	486	604	700	801	950	920	1018

81 It will be seen, however, from column 8, that $\log \frac{\theta_s - \theta_1}{\theta_s - \theta_2}$ decreased regularly from the first to the last section. Column 9 gives the absolute film temperature as calculated from formula [43]. From this it will be seen that $\log \frac{\theta_s - \theta_1}{\theta_s - \theta_2}$, while varying inversely with the absolute film temperature, varies at a much less rapid rate. Column 12 gives the value of K, the factor of transmission, as calculated from the values in column 8 according to formula [36]. From the experimental constants, determined in the manner to be de-

scribed later, we are able to calculate the variations in the value of R, (according to formula [29]) as shown in column 13, from the values of K as determined from the tests. In column 10 the variations in R have been calculated on the assumption that the film resistance is directly proportional to the absolute film temperature. The sub-

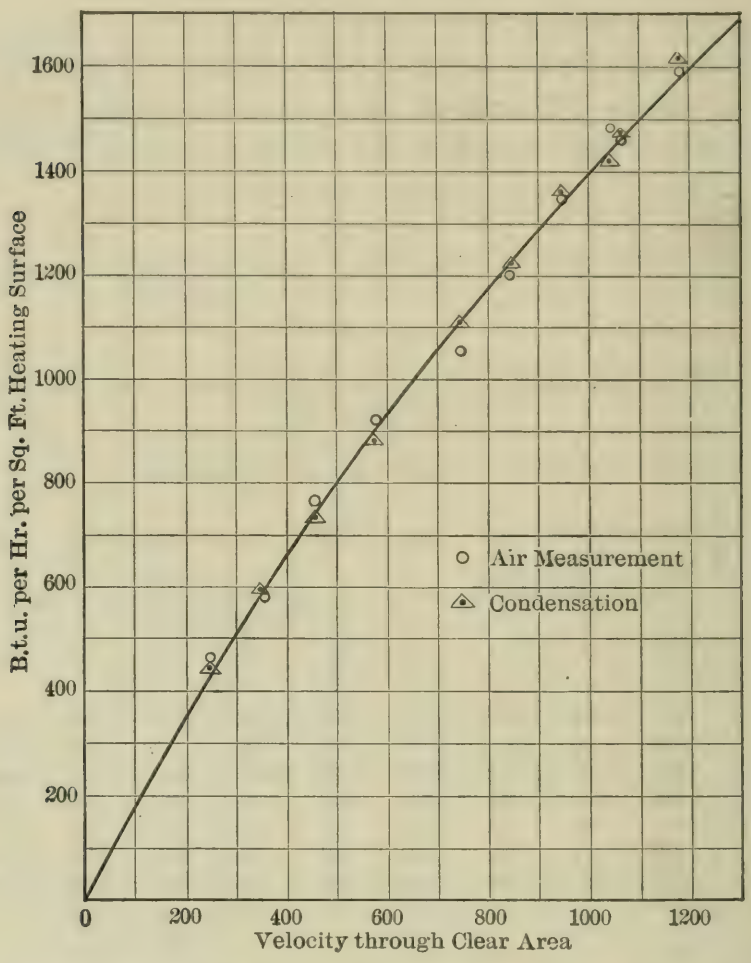


FIG. 33 COMBINED CURVE SHOWING RATE OF TRANSMISSION FROM CONDENSATION AND AIR MEASUREMENT CORRECTED FOR 8 PER CENT ERROR IN AIR MEASUREMENTS

stantial agreement of the values thus calculated, with the values of R in column 13, as calculated from the tests, is thus conclusively demonstrated. In column 11 the value of K has been calculated from formula [29], using the values of R as calculated from the film temperature. The close agreement of these values of K as calculated from the formula may be shown by comparison with the values of K as determined from tests in column 12.

82 The same substantial agreement having been verified at other velocities, we are warranted in asserting that the value of R varies directly as the calculated absolute film temperature, where R is the resistance of the surface film to the transmission of heat. The value of R as used in these calculations includes not only the resistance of the surface film of air, but also the surface film of steam and of the conducting wall. However, as has been proven by experiments in heat transmission from steam to water (see Fig. 28), this resistance is less than one per cent of the resistance of the air film. Therefore

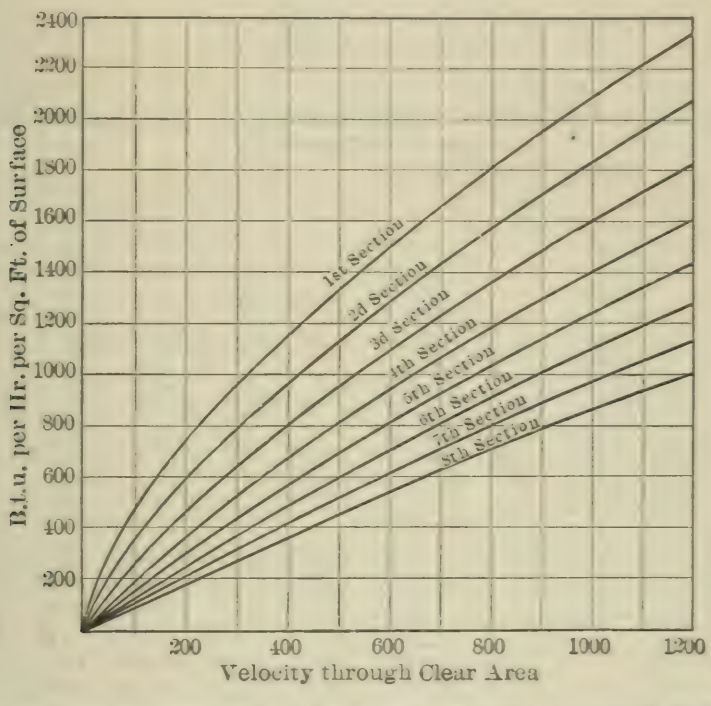


FIG. 34 RATE OF TRANSMISSION IN CONSECUTIVE SECTIONS OF HEATER AT VARIOUS VELOCITIES

we may take the entire resistance R as varying directly as the absolute film temperature, with negligible error.

DETERMINATION OF THE VALUES *R* AND *B*

83 As shown in the preceding paragraph, the absolute values of R are dependent upon the film temperature, but in order to calculate this film temperature we must first know the approximate values of both R and K. These values were first determined approximately for different velocities by assuming R to be constant, that is K to be constant, for any fixed velocity. K may then be calculated directly from the values in Fig. 33 by means of equation [36], in which

$$K = \frac{C_p G}{S} \log_e \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2} \right) \quad \ldots\ldots\ldots\ldots\ldots\ldots [46]$$

where

C_p = the specific heat of air

G = the weight of air in lb. per hour passed through the heater

S = the total sq. ft. of heating surface

84 From equation [29] we have

$$K = \frac{1}{R + \frac{B}{C_p W_o V_c}}$$

TABLE 10 SHOWING THE FILM RESISTANCE R TO VARY AS THE ABSOLUTE FILM TEMPERATURE

VELOCITY THROUGH CLEAR AREA 1000 FT. PER MIN. θ_s = 227 DEG. FAHR. θ_1 = 20 Deg. Fahr.

1	2	3	4	5	6	7	8	9	10	11	12	13
Number of Section	B.t.u. per Sq. Ft. per Section	Temperature Rise per Section	Temperature of Air Leaving Section	Average Absolute Temperature in Each Section θ_A	Final Temperature Difference $\theta_s - \theta_2$	Ratio of Leaving to Entering Temperature Difference $\frac{\theta_s - \theta_2}{\theta_s - \theta_1}$	$\frac{\theta_s - \theta_1}{\text{Log}\ \frac{\theta_s - \theta_1}{\theta_s - \theta_2}}$	Absolute Film Temperature θ'	R Calculated from Film Temperature	K Calculated from Formula	K from Test	R Calculated from Test
1	2092	23.75	43.75	491.88	183.25	0.88528	0.05292	596.60	0.04267	10.72	10.74	0.04250
2	1838	20.88	64.63	535.07	162.37	0.88605	0.05254	615.78	0.04404	10.56	10.64	0.04333
3	1614	18.32	82.95	552.10	144.05	0.88725	0.05195	623.77	0.04461	10.55	10.55	0.04414
4	1418	16.10	99.05	567.10	127.95	0.88830	0.05144	630.79	0.04611	10.44	10.52	0.04438
5	1249	14.18	113.23	580.32	113.77	0.88910	0.05105	636.99	0.04555	10.39	10.36	0.04587
6	1104	12.53	125.76	592.03	101.24	0.88920	0.05100	642.48	0.04595	10.35	10.38	0.04565
7	981	11.14	136.90	602.47	90.10	0.89000	0.05061	647.37	0.04630	10.32	10.27	0.04668
8	878	19.97	146.87	611.86	80.13	0.88935	0.05093	651.77	0.04661	10.29	10.33	0.04615

from which we may deduce the resistance formula

$$\frac{1}{K} = R + \left(\frac{B}{C_p W_o} \right) \frac{1}{V_o} \quad \ldots\ldots\ldots\ldots\ldots\ldots [47]$$

in which R is assumed to be a constant, B is a constant to be determined, $C_p W_o$ is a known constant, $\frac{1}{V_o}$ is a known variable, and $\frac{1}{K}$ is the reciprocal of K as calculated above, in equation [46].

85 By inspecting equation [47] we see that, if R is assumed to be a constant, this is a straight line relationship. Therefore the approxi-

mate values of R and $\frac{B}{C_p W_o}$ may be determined from the equation of a straight line drawn through the values of $\frac{1}{K_o}$ plotted to the corresponding values of $\frac{1}{V_o}$. Having determined the values in this manner we may now determine approximately the values of the mean absolute film temperatures at the different velocities using the average rates of transmission as shown in Fig. 33.

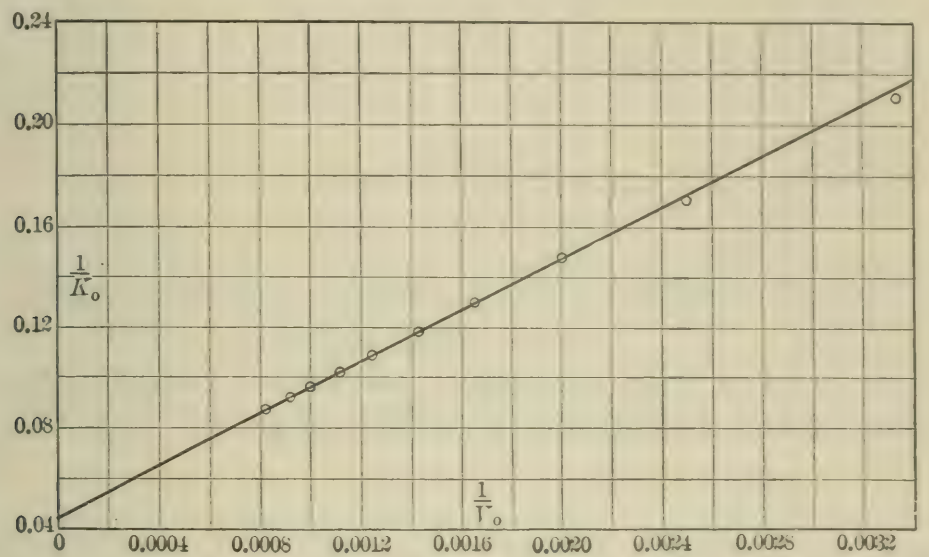


FIG. 35 CURVE SHOWING DERIVATION OF CONSTANTS IN RATIONAL CONVECTION FORMULA

$$\frac{1}{K_o} = R_o + \left(\frac{B}{C_p W_o}\right)\left(\frac{1}{V_o}\right) \quad \text{or} \quad K_o = \frac{1}{R_o + \frac{B}{C_p W_o V_o}}$$

86 The mean absolute film temperature may be determined by the formula

$$\theta'_m = \theta_s - RK(\theta_s - \theta)_m = \theta_s - RK \frac{(\theta_2 - \theta_1)}{\log_e \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right)} \quad \ldots\ldots [48]$$

for the derivation of which see Appendix No. 2, where

θ'_m = mean absolute film temperature

$(\theta_s - \theta)_m$ = mean temperature difference between the steam and air

87 For eight sections of heater with the velocity of the air through the clear area of the heater, measured at 70 deg. fahr. at 1000 ft. per minute, we find a mean absolute film temperature of approximately

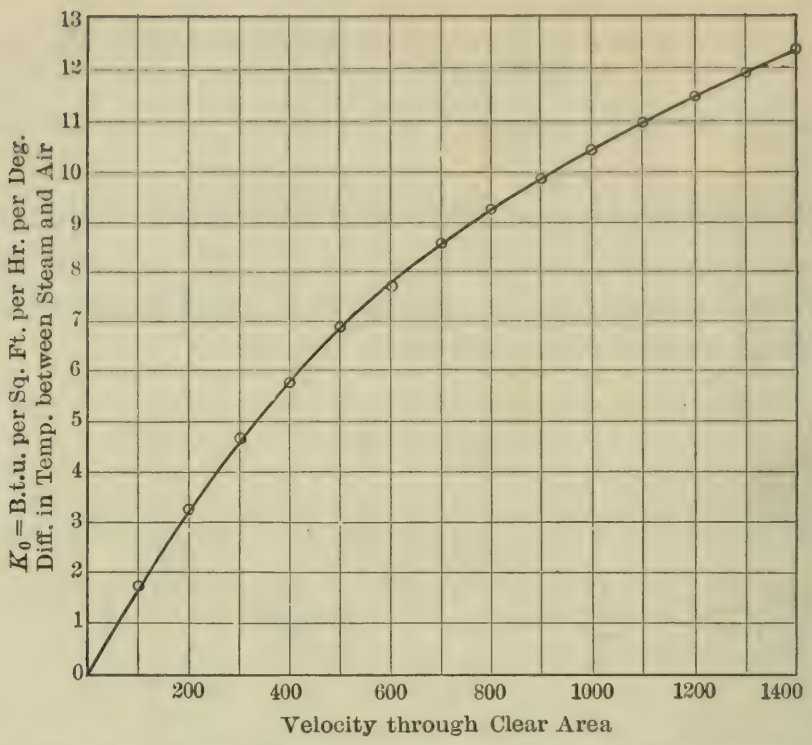


FIG. 36 RATE OF HEAT TRANSMISSION FOR VARIOUS VELOCITIES, VALUES OF K_0 FROM TEST CORRECTED TO ABSOLUTE FILM TEMPERATURE OF 625 DEG.

TABLE 11 COEFFICIENTS OF HEAT TRANSMISSION AT VARIOUS VELOCITIES

1	2	3	4	5	6	7	8	9	10	11	12	13
V	$\frac{1}{\sqrt{V}}$	$\frac{H}{S}$ = B.t.u. per Sq. Ft.	$\theta_3-\theta_1=\frac{8\times\frac{H}{S}}{\frac{A\times 60\times V}{55.22\times S}}$	$\theta_3-\theta_1$	K	$\frac{1}{K}$	$(\theta_s-\theta)$m	$K(\theta_s-\theta)$m	X	$\frac{1}{K_0}=\frac{1}{K}-X$	K_0 from Test	K_0 Calculated from Formula
200	0.0050	355	160.8	46.2	3.316	0.3015	107.0	355	0.00331	0.29819	3.354	3.356
300	0.00333	518	156.5	50.5	4.674	0.2140	111.0	520	0.00279	0.21121	4.735	4.689
400	0.0025	667	151.1	55.9	5.788	0.1730	115.4	667	0.002295	0.17071	5.858	5.838
500	0.0020	805	145.9	61.1	6.743	0.1480	119.6	806	0.001815	0.14619	6.841	6.845
600	0.001667	936	141.3	65.7	7.614	0.1310	123.0	938	0.00143	0.12957	7.718	7.746
700	0.00143	1060	137.2	69.8	8.412	0.1190	126.5	1067	0.001012	0.11799	8.476	8.540
800	0.00125	1178	133.4	73.6	9.146	0.1093	129.0	1180	0.000639	0.10866	9.204	9.251
900	0.00111	1290	129.9	77.2	9.823	0.1018	131.3	1290	0.000279	0.10152	9.850	9.896
1000	0.00100	1398	126.7	80.3	10.465	0.0956	133.5	1397	−0.0000585	0.09566	10.455	10.490
1100	0.000909	1500	123.5	83.5	11.040	0.0905	136.0	1507	−0.000418	0.09092	11.000	10.920
1200	0.000833	1599	120.8	86.3	11.610	0.0861	137.7	1600	−0.000936	0.08704	11.490	11.400

625 deg. Therefore we have calculated the value of R_o and all values of K_o at different velocities for this assumed standard of film temperature. Having calculated the average values of K for eight sections of heater and different velocities as described in the preceding paragraph, we may determine the value of K_o for a film temperature of 625 deg. from the formula.

$$K_o = K\left[1 + K_o R_o\left(\frac{\theta'_m - 625}{625}\right)\right] \quad \ldots\ldots\ldots\ldots [49]$$

for derivation of which see Appendix No. 3.

88 These values as calculated from the tests are shown in column 12 of Table 11. In this table

$$X = R_o\left(\frac{\theta'_m - 625}{625}\right)$$

We now have the rates of transmission for different velocities corrected to a standard absolute film temperature of 625 deg. Therefore, if our theoretical relationships hold true

$$\frac{1}{K_o} = R_o + \left(\frac{B}{C_p W_o}\right)\frac{1}{V_o}$$

is the equation of a straight line, in which R_o is the intercept at $\frac{1}{V_o} = 0$ and $\frac{B}{C_p W_o}$ is the slope. The plot of these values for different velocities as determined from the test curve Fig. 33 is shown in Fig. 35. It will be noticed from an inspection of Fig. 35 that these points do lie in a straight line, with remarkable accuracy, thus showing the agreement of the test results with the rational formula

$$K_o = \frac{1}{R_o + \frac{B}{C_p W_o V_o}}$$

89 Substituting numerical values obtained from this plot we have

$$\frac{1}{K_o} = 0.0447 + \frac{50.66}{V_o} \quad \ldots\ldots\ldots\ldots\ldots [50]$$

Hence

$$K_o = \frac{1}{0.0447 + \frac{50.66}{V_o}} \quad \ldots\ldots\ldots\ldots\ldots [51]$$

Column 13, Table 11, shows the value of K_o calculated from this formula, which may be compared with the values of K_o calculated from test and given in column 12.

90 Fig. 36 gives the value of K_o for different velocities, the curve being plotted from the formula and the points shown being the values determined directly from the test.

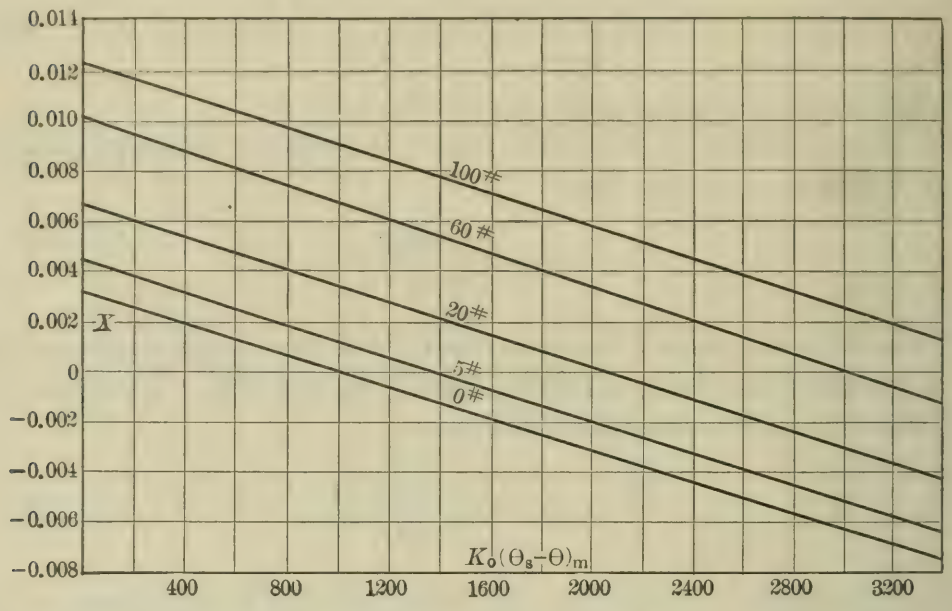


FIG. 37 RELATION BETWEEN $K_o(\theta_s - \theta)_m$ AND X AT DIFFERENT STEAM PRESSURES

RATIONAL FORMULA FOR HEAT TRANSMISSION

91 We may now insert the above numerical values given in equation [51] in the rational formulae [38] to [41]. Substituting in equation [40] we have

$$\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)=\frac{S}{0.1119Q+127A} \quad \ldots\ldots\ldots\ldots [52]$$

Substituting proper values in equation [41] we have

$$\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)=\frac{f}{0.1119V+127} \quad \ldots\ldots\ldots\ldots [53]$$

also

$$\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)=\frac{0.3994fK}{V} \quad \ldots\ldots\ldots\ldots\ldots [54]$$

where

θ_s = steam temperature

θ_1 = entering air temperature

θ_2 = final air temperature

$f = \frac{S}{A}$ = ratio of total surface to clear area

V = the velocity of air through clear area (at 70 deg.) in ft. per min.

Q = cu. ft. air per min. at 70 deg. fahr.

TABLE 12 CALCULATIONS FOR CORRECT VALUES OF FINAL TEMPERATURES AND HEAT TRANSMISSION

VELOCITY THROUGH CLEAR AREA 1000 FT. PER MIN. θ_S = 227 DEG. FAHR. θ_1 = 20 DEG. FAHR.

1	2	3	4	5	6	7	8	9	10	11	12
Number of Section	f	Log of Approximate $\frac{\theta_s-\theta_1}{\theta_s-\theta_2}$	$K_o X$ from Curve	$\frac{\text{Log}\frac{\theta_s-\theta_1}{\theta_s-\theta_2}}{1+K_o X}$	$\frac{\theta_s-\theta_1}{\theta_s-\theta_2}$ Corrected	$\theta_S \quad \theta_2$	Final Temperature	Total Temperature Rise	Average B.t.u. per Sq. Ft., Total	Temperature Increment in Section	Average B.t.u. per Sq. Ft. per Section
1	12.335	0.05167	−0.0224	0.05285	1.130	183.20	43.80	23.80	2096.4	23.80	2096.5
2	24.67	0.10334	−0.0188	0.10534	1 2745	162.42	64.58	44.58	1963.5	20.78	1830.4
3	37.005	0.15501	−0.0152	0.15739	1 4368	144.07	82.93	62.93	1840.8	18.35	1616.3
4	49.34	0.20668	−0.0115	0.20909	1.6174	127.98	99.02	79.02	1740.0	16.09	1417.2
5	61.675	0.25835	−0.0084	0.26052	1.8220	113.61	113.39	93.39	1645.5	14.37	1265.8
6	74 01	0.31002	−0.00533	0.31168	2.0497	101.00	126.00	106.00	1556.0	12.61	1110.8
7	86.345	0.36169	−0.00225	0.36250	2.304	89.85	137.15	117.15	1474.2	11.15	982.2
8	98.68	0.41336	+0.00084	0.41301	2.5883	79.98	147.02	127.02	1398.5	9.87	869.4

92 The above formulae are correct for a mean film temperature of 625 deg. absolute. For approximate results, these formulae can be used without corrections. For more accurate results the factor 0.1119 must be corrected for the calculated absolute film temperature in each case. The full formula embodying this correction, where extreme accuracy is desired, is

$$f = (0.0001791 V_o \theta_s + 126.8) \cdot \log_{10}\left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) - \frac{0.000003474\, V_o^2 (\theta_2 - \theta_1)}{0.0447\, V_o + 50.66} \quad [55]^*$$

which can only be solved for f when the initial and final temperature of the air are known. For ordinary calculations where we wish to

*See Appendix No. 4 for derivation.

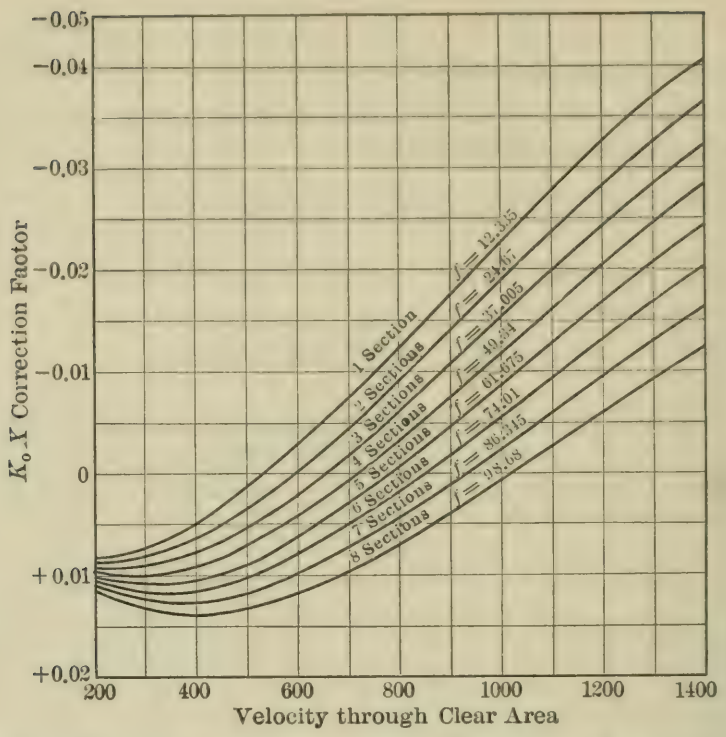


FIG. 38 CORRECTION FACTORS FOR COEFFICIENT OF HEAT TRANSMISSION

$$K = \frac{K_o}{1 + K_o X} \qquad \theta_s = 227 \text{ deg. fahr.} \qquad \theta_1 = 20 \text{ deg. fahr.}$$

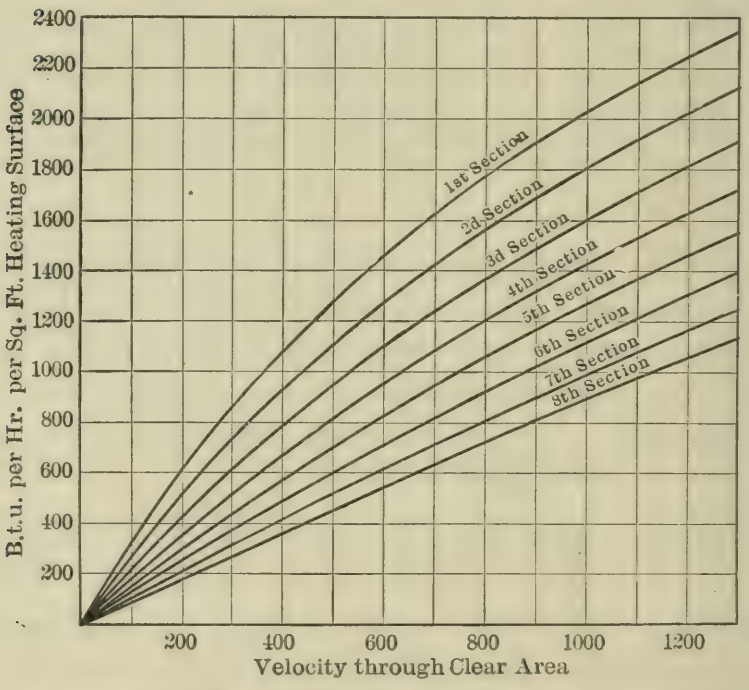


FIG. 39 APPROXIMATE CALCULATED VALUES OF HEAT TRANSMISSION

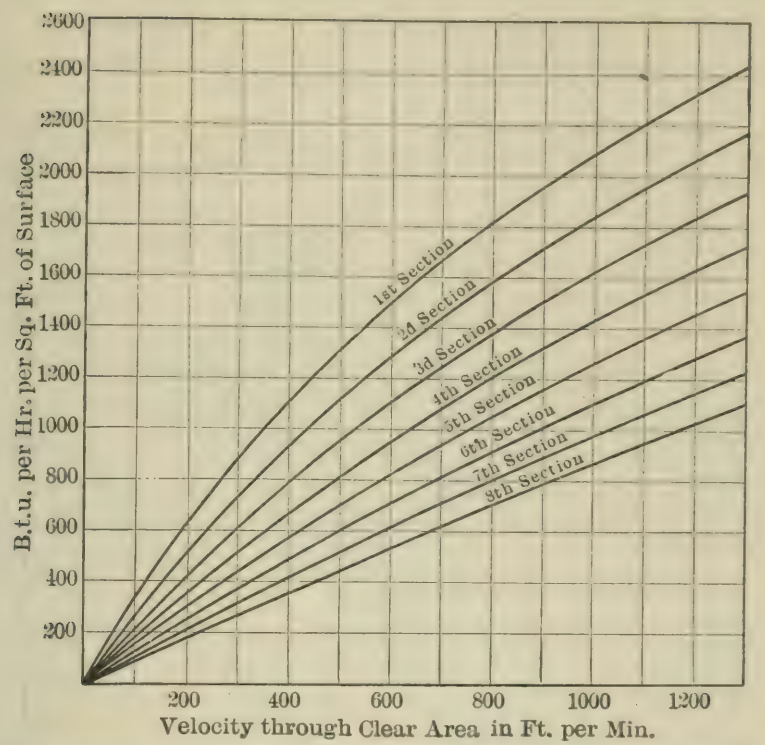


FIG. 40 CORRECTED CALCULATED VALUES OF HEAT TRANSMISSION

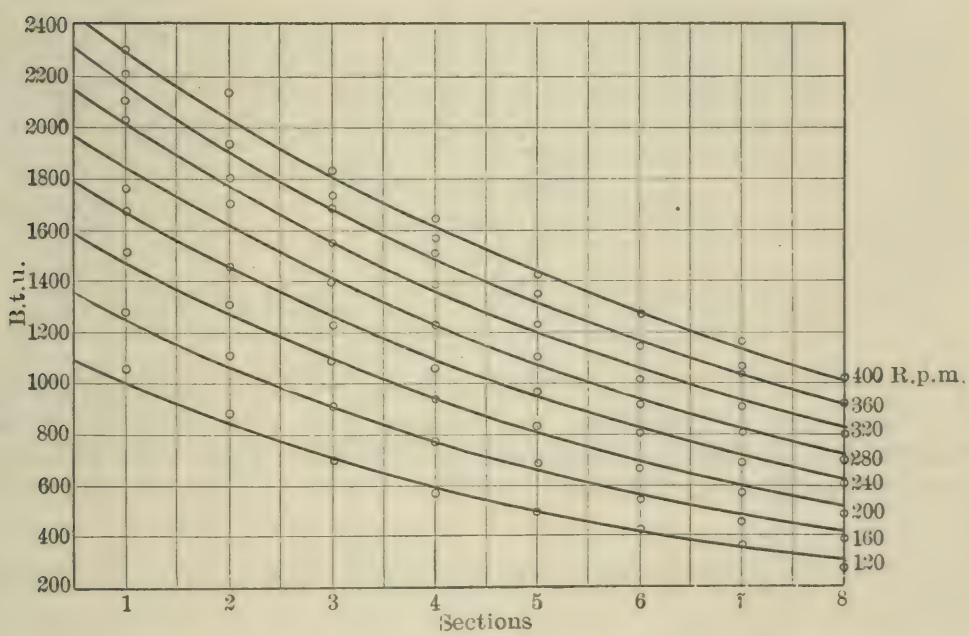
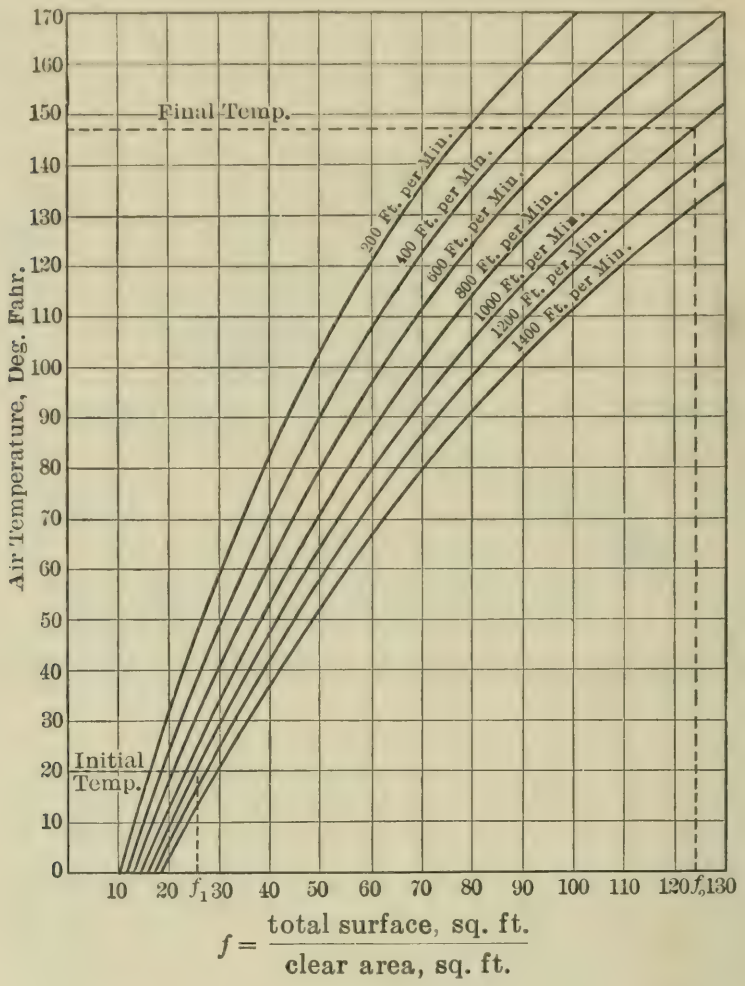


FIG. 41 COMPARISON OF CALCULATED VALUES WITH TEST RESULTS. LINES REPRESENT CALCULATED VALUES; POINTS REPRESENT TEST RESULTS

solve for the initial and final temperatures where f and V are known, we first determine approximate values of θ_1 and θ_2 from equation [53]. We then solve for the mean temperature difference between



or

$$f = (0.000179\, V_0 \theta_s + 126.8) \log_{10} \frac{\theta_s - \theta_1}{\theta_s - \theta_2} - (0.000003474\, V_0 K_0)(\theta_2 - \theta_1)$$

FIG. 42 RELATION BETWEEN HEATER SURFACE AND TEMPERATURES OF THE AIR AT VARIOUS VELOCITIES FOR WROUGHT-IRON PIPE HEATERS. STEAM PRESSURE = 5 LB.; STEAM TEMPERATURE = 227 DEG. FAHR.; AIR VELOCITY MEASURED AT 70 DEG. FAHR.

steam and air, using the above determined values of θ_1 and θ_2 in the formula

$$(\theta_s - \theta)_m = \frac{0.434\,(\theta_2 - \theta_1)}{\log_{10}\left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right)} \text{ (see Appendix No. 2)} \quad [56]$$

93 Having determined $(\theta_s - \theta)_m$, we may then determine the value of X from the formula

$$X = 0.00007152\theta_s - 0.0447 - 0.0000032\, K_o\, (\theta_s - \theta)_m \ldots . [57]$$

as derived from Appendix No. 3, or from Fig. 37 giving the values of X for the various values of $K_o(\theta_s - \theta)_m$. The correct value for θ_2 will be given by the equation

$$\log_{10}\left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) = \frac{0.3994\, f K_o}{V(1 + K_o X)} \text{ (see Appendix No. 3) } \ldots . [58]$$

To simplify this correction for a series of calculations, curves may be constructed for the correction factor as shown in Fig. 38.

94 An example of such a calculation is given in Table 12. This should be compared with the results from tests given in Table 10. for the same conditions. Comparison of the following corresponding columns in each table is of interest:

	TABLE 10		TABLE 12
	Test Values		Calculated Values
Compare column	2	with	12
Compare column	3	with	11
Compare column	4	with	8
Compare column	6	with	7

95 Fig. 39 shows the approximate calculated values of heat transmission in the individual sections at various velocities, without correction for film temperature. Fig. 40 shows the same values corrected for film temperature. It is interesting to compare these results with Fig. 34, in which the corresponding test values are given. The agreement of the actual test results with the corresponding calculated values is still more strikingly shown in Fig. 41, in which the curves are accurately drawn in accordance with the corrected formula, while the plotted points represent the actual test values given in Table 8.

96 Fig. 42 shows graphically the relation between the heater surface and air temperature for various velocities through wrought-iron pipes heaters. The factor f is the ratio of total surface to clear area and will vary with different number of heater sections, being equal to 98.68 for the eight sections tested. The values of f as plotted were computed from formula [7], Appendix No. 4.

97 As an example of the use of Fig. 42, we will assume an initial or entering air temperature of 20 deg. fahr. with a velocity through

the clear area of 1000 ft. per min. Following the dotted line as shown, we find a value of $f_1 = 25.4$, and adding $f = 98.68$ for eight sections gives $f_2 = 124.08$. From $f_2 = 124.08$ follow the dotted line upward till it intersects the 1000 ft. per min. velocity and the across to a final or leaving temperature of 147 deg. fahr. An inspection of column 8 of Table 12 shows a final temperature obtained from the tests of 147.02 deg. fahr. This diagram may be used for any other value of temperature or surface, when the steam pressure is 5 lb. gage.

APPENDIX No. 1

98 A series of ten tests was made in February 1906 for the purpose of checking the results obtained from similar tests made in 1902. As the results of these tests agreed so fully with those of the former tests, it was considered unnecessary to work them up in detail until the present time. A number of similar tests were also made at this time at various other steam pressures. As these merely substantiate the relationships here presented, it has been thought unnecessary to include them.

99 The tests were made on an indirect or fan system heater, consisting of eight sections each of four rows of 1-in. wrought-iron pipe. Details of the arrangement of the pipes and their distribution in the cast-iron base is shown in Fig. 30. Each section contained 126.6 sq. ft. of surface, and the clear area was 10.263 sq. ft. The air was drawn through the heater by a steel plate fan, and blown through a nozzle where the air measurements were made. The velocity of the air was measured by means of a pitot tube and a hook gage, the temperatures being carefully taken at the outlets with thermometers well protected from the effect of radiation. The pitot tube readings were taken at the outlet of the fan nozzle, a series of 64 readings having been taken over the area of the outlet, and the ratio of the average to the reading at the center carefully computed. During the test continuous readings were taken with the pitot tube fixed in the center of the orifice, which were afterwards corrected for the above ratio.

100 The pressure of the steam, as well as the speed of the fan engine, was kept constant throughout each test by two of the observers. The condensation from each section was weighed separately, and the temperatures carefully noted. Each section was provided with an air vent, and thoroughly blown out for about one hour before starting the test, and some steam was allowed to blow through during the entire test. A separator was placed in the steam line before the throttle, and a calorimeter was used to determine the quality of the steam as it entered the steam heater. A slight degree of superheat was observed during each test, and the proper allowance made in the calculation of the total heat of the steam. The saturation temperature of the steam was also taken before and after each test to check the steam gage, proper allowance being made for stem correction to the thermometer. Barometric readings were made during each test, and the steam pressure so regulated as to maintain an absolute pressure of 19.7 lb. on the heating coils. The steam gage was calibrated by means of a dead-weight tester before and after each test, due allowance being made for the barometric reading taken at the time.

101 In Table 8 will be found the principal data obtained. As already explained, the first column for each test contains the uncorrected observations, the second column these same readings corrected to standard conditions of 70 deg. fahr. and 29.92 in. barometer, while in the third column, as previously explained, a further correction is made for a systematic error of 8 per cent in

measuring the air. Proper allowance had been made for heat loss from the air by convection from the surface of the heater casing and fan housing, correction for this loss being added to the air temperature. Slight corrections were also made for condensation from such portions of the heater base and pipe connections as were exposed to the air of the room rather than to the air measured. This loss was found to amount to 78.4 B.t.u per hour per degree difference between the steam temperature and room temperature.

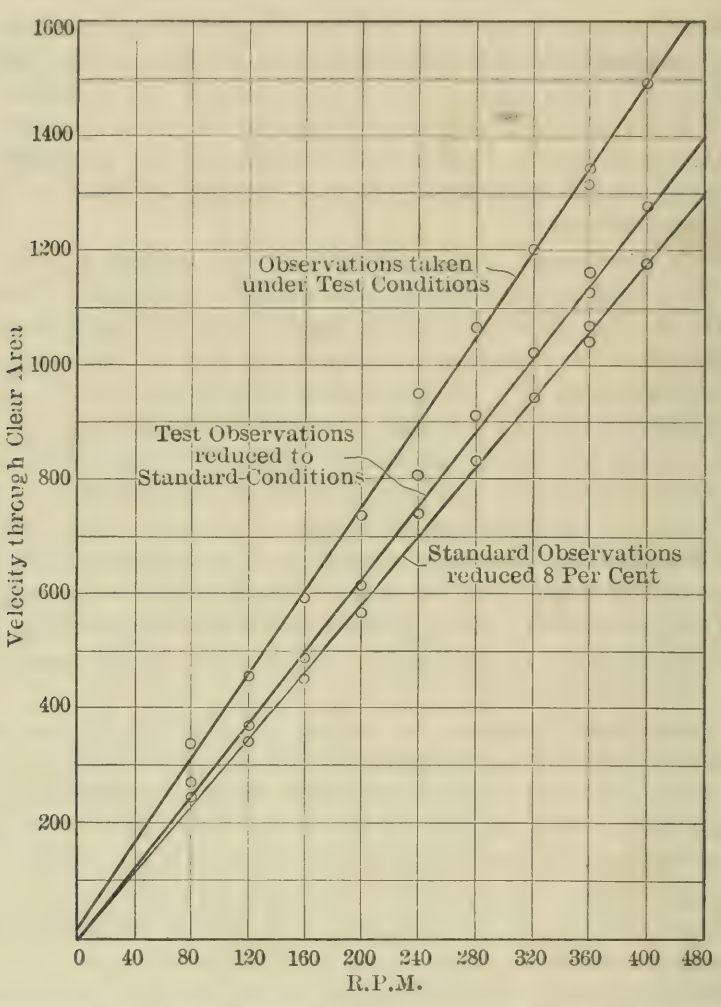


FIG. 43 RELATION OF FAN SPEED TO VELOCITY OF AIR THROUGH HEATER

102 A further correction was also required to make allowance for the fact that the condensation left the heater at practically steam temperature, and a portion would flash into steam upon exposure to the atmosphere. After careful tests it was decided to add 1.5 per cent to the weight of condensation to cover this loss, as well as 1½ lb. per ½ hr. to cover the average loss due to evaporation from the tanks containing the condensation. A final correction was then made to reduce all the tests to a standard of 20 deg. for the temperature of the entering air. This was based on the fact that at any velocity the weight of the condensation practically varies directly as the temperature difference between the steam and entering air.

103 As previously stated, the speed of the fan was maintained constant during each test at the speeds shown in Table 8. The relation of the fan speed to the velocity through the clear area as computed from the air measurements, is shown in Fig. 43 plotted from the values in Table 8, barometric measurements being taken into consideration in computing the air measurements. It will be noted that the velocities recorded are fairly regular, with the exception of the tests made at 80 and 240 r.p.m.

APPENDIX No. 2

FORMULA FOR MEAN EFFECTIVE TEMPERATURE

104 Let H be the total heat transferred, S the total surface, K the rate of heat transmission per degree difference in temperature, $(\theta_s-\theta)_m$ the mean effective temperature difference between the steam and air. Then we will have

$$H=K(\theta_s-\theta)_m S \quad \ldots\ldots [1]$$

$$(\theta_s-\theta)_m=\frac{H}{KS}=\frac{C_p W(\theta_1-\theta_2)}{KS} \quad \ldots\ldots [2]$$

also

$$dH=(\theta_s-\theta)\,K dS \quad \ldots\ldots [3]$$

105 Referring to Fig. 29 we have

$$dH=-C_p W d(\theta_s-\theta) \quad \ldots\ldots [4]$$

where W is the total pounds of air per hour passing over the surface S.

106 Equating [3] and [4] we have

$$(\theta_s-\theta)K dS=-C_p W d(\theta_s-\theta) \quad \ldots\ldots [5]$$

$$K\int_0^s dS=C_p W\int_{\theta_s-\theta_2}^{\theta_s-\theta_1}-\frac{d(\theta_s-\theta)}{\theta_s-\theta} \quad \ldots\ldots [6]$$

Integrating

$$KS=C_p W\log_e\frac{\theta_s-\theta_1}{\theta_s-\theta_2} \quad \ldots\ldots [7]$$

Hence

$$(\theta_s-\theta)_m=\frac{\theta_2-\theta_1}{\log_e\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)} \quad \ldots\ldots [8]$$

$$(\theta_s-\theta)_m=\frac{(\theta_2-\theta_1)-(\theta_s-\theta_2)}{\log_e\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)} \quad \ldots\ldots [9]$$

107 From equation [45] we have

$$\theta_s-\theta'=RK(\theta_s-\theta)$$

or approximately

$$\theta_s-\theta'=R_o K_o(\theta_s-\theta) \quad \ldots\ldots [10]$$

Hence

$$d(\theta_s - \theta') = RKd(\theta_s - \theta) \quad \ldots\ldots [11]$$

Therefore it follows from [10] and [11] that we will have the same relationship between $(\theta_s - \theta')$, $(\theta_s - \theta'_2)$ and $(\theta_s - \theta')_m$ as between $(\theta_s - \theta_1)$, $(\theta_s - \theta_2)$ and $(\theta_s - \theta)_m$ in equation [11] Hence

$$(\theta_s - \theta')_m = \frac{(\theta_s - \theta'_1)(\theta_s - \theta'_2)}{\log_e \left(\frac{\theta_s - \theta'_1}{\theta_s - \theta'_2}\right)} \quad \ldots\ldots [12]$$

108 Substituting in [12] $R_o K_o(\theta_s - \theta_1)$ for $\theta_s - \theta'_1$ and $R_o K_o(\theta_s - \theta_2)$ for $\theta_2 - \theta'_2$ we have

$$(\theta_s - \theta')_m = \frac{R_o K_o[(\theta_s - \theta_1) - (\theta_s - \theta_2)]}{\log_e \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right)} \quad \ldots\ldots [13]$$

or

$$(\theta_s - \theta')_m = \frac{R_o K_o(\theta_2 - \theta_1)}{\log_e \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right)} \quad \ldots\ldots [14]$$

or

$$\theta'_m = \theta_s - \frac{R_o K_o(\theta_2 - \theta_1)}{\log_e \left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right)} \quad \ldots\ldots [15]$$

APPENDIX No. 3

PROOF OF FORMULAE [49] AND [58]

$$K_o = K\left[1 + K_o R_o\left(\frac{\theta'_m - 625}{625}\right)\right] \quad \text{.....[49]}$$

109 It has been shown that the value of R for a mean film temperature of 625 deg. is 0.0447, and that it varies directly as the absolute temperature. Hence

$$R = \frac{R_o\,\theta'_m}{625} = \frac{0.0447\,\theta'_m}{625} \quad \text{.....[1]}$$

Hence

$$\frac{1}{K} = \frac{R_o\,\theta'_m}{625} + \frac{50.66}{V} \quad \text{.....[2]}$$

$$= \frac{1}{K_o} + R_o\left(\frac{\theta'_m - 625}{625}\right) \quad \text{.....[3]}$$

$$K_o = K + KK_oR_o\left(\frac{\theta'_m - 625}{625}\right) \quad \text{.....[4]}$$

$$= K\left[1 + K_oR_o\left(\frac{\theta'_m - 625}{625}\right)\right] \quad \text{.....[5]}$$

110 Let

$$\mathrm{X} = R_o\left(\frac{\theta'_m - 625}{625}\right) \quad \text{.....[6]}$$

Then

$$K_o = K(1 + \mathrm{X}K_o) \quad \text{.....[7]}$$

$$K = \frac{K_o}{1 + \mathrm{X}K_o} \quad \text{.....[8]}$$

111 We have from equation [54]

$$\log_{10}\left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) = \frac{0.3994\,f\,K}{V} \quad \text{.....[9]}$$

Hence substituting the above value of K we have

$$\log\left(\frac{\theta_s - \theta_1}{\theta_s - \theta_2}\right) = \frac{0.3994 f K_o}{V(1 + K_o\mathrm{X})} \quad \text{.....[10]}$$

112 From equation [6] we have $X = R\left(\frac{\theta'_m - 625}{625}\right)$. From equation [48] we have $\theta'_m = \theta_s - RK(\theta_s - \theta)_m$. Hence

$$\theta'_m = \theta_s - R_o K_o (\theta_s - \theta)_m \text{ (approximately)} \quad \ldots\ldots\ldots\ldots [11]$$

Substituting this value of θ'_m in equation [6] we have

$$X = R_o \left[\frac{\theta_s - R_o K_o (\theta_s - \theta)_m - 625}{625}\right] \quad \ldots\ldots\ldots\ldots\ldots [12]$$

113 Substituting numerical values previously determined

$$X = 0.0447 \left[\frac{\theta_s - 0.0447\, K_o (\theta_s - \theta)_m - 625}{625}\right] \quad \ldots\ldots\ldots\ldots [13]$$

Hence

$$X = 0.00007152\,\theta_s - 0.0447 - 0.0000032\, K_o (\theta_s - \theta)_m \quad \ldots\ldots [14]$$

APPENDIX No. 4

PROOF OF EQUATION [55]

114 From equation [54] we have

$$\log\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)=\frac{0.3994\,f\,K}{V} \quad \ldots\ldots[1]$$

and from equation [2], Appendix No. 3

$$\frac{1}{K}=\frac{0.0447\,\theta'_m}{625}+\frac{50.66}{V} \quad \ldots\ldots[2]$$

115 From equation [15] Appendix No. 2

$$\theta'_m=\theta_s-0.434\times 0.0447\,K_o\left(\frac{\theta_2-\theta_1}{\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)}\right) \quad \ldots\ldots[3]$$

Hence

$$\frac{1}{K}=\frac{0.0447\,\theta_s}{625}-\frac{0.0447\times 0.434\times 0.0447}{625}\,K_o\left(\frac{\theta_2-\theta_1}{\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_1}\right)}\right)+\frac{50.66}{V} \quad \ldots\ldots[4]$$

$$=0.00007152\,\theta_s-0.0000013875\,K_o\left(\frac{\theta_2-\theta_1}{\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)}\right)+\frac{50.66}{V} \quad \ldots\ldots[5]$$

$$\frac{0.3994\,f}{V}=\left(0.00007152\,\theta_s+\frac{50.66}{V}\right)\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)-0.0000013875\,K_o(\theta_2-\theta_1) \quad \ldots[6]$$

or

$$f=(0.0001791\,V_o\,\theta_s+126.8)\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)-(0.000003474\,V_o\,K_o)\,(\theta_2-\theta_1) \quad \ldots[7]$$

$$=(0.0001791\,V_o\,\theta_s+126.8)\log_{10}\left(\frac{\theta_s-\theta_1}{\theta_s-\theta_2}\right)-\frac{0.000003474\,V_o^2\,(\theta_2-\theta_1)}{0.0447\,V_o+50.66} \quad \ldots\ldots[8]$$

DISCUSSION

R. C. Carpenter said that there was a great deal of interest in that part of the paper pertaining to the apparatus for obtaining and maintaining uniform hygrometric conditions. The theoretical formula for the effect of convection at different velocities of air from different radiators is a remarkable contribution.

He conducted an extended series of experiments[1] several years ago for obtaining the heat transmission for different air velocities and for different amounts of surfaces, in which a fan blower was employed to deliver air over various amounts of surfaces. The results obtained in these experiments were entirely the effect of convection. The general results of these experiments indicated that the convected heat transmitted per degree difference of temperature per sq. foot of heating surface could be expressed quite accurately by a formula consisting of a constant part and a part which varies with the square root of the velocity of the air. Thus if r = heat of transmission per degree difference of temperature, a and b = constants, v = velocity of the air in feet per second, then $r = a + b\sqrt{v}$. For the ordinary conditions of temperature which prevail in heating with fans, $a = 1.25$ and $b = 0.6$.

He was pleased to note the attempt which Mr. Carrier has made to express this character of heat transmission in a rational manner.

F. R. Still (written). The advantages of the proper conditioning of air are well known to engineers. Recent investigations in Germany indicate that even very impure air, if properly humidified, is less harmful to human beings than very dry air of a high degree of purity. In respect to disease, it was discovered some years ago that germs are carried from one person to another on small particles of dust and are breathed into the nose, throat and lungs. With the modern system of air conditioning the dust is almost entirely eliminated which lessens to a high degree the probability of germs passing from one person to

[1] Heating and Ventilating Buildings.

another. On the score of economy also, there is a saving in heating, as a temperature of 66 deg., with a proper degree of humidity is more comfortable than a very dry atmosphere when the temperature is considerably above 70 deg.

The devices now on the market are all intended for public buildings, such as schools, theatres, churches, etc.; but what is wanted most of all is a small, cheap and effective device by means of which the humidity in our residences can be controlled.

J. I. Lyle (written). With a view to supplementing what Mr. Carrier has given, it probably is not out of place to call attention to the fact that the latest designs for dry blast plants are making use of this direct spray system. Usually water is employed in the first stage of work, which can often be obtained from natural sources sufficiently cold to do a considerable portion of the work. Even where the water has to be cooled by refrigeration, a very high ammonia suction pressure can be used, thereby greatly reducing the size of ammonia compressor required and horsepower consumed.

If the dehumidifying is being done on the suction side of the blowing engine, and much less than $2\frac{1}{2}$ grains of water vapor, corresponding to a dewpoint of 36 deg., is to be carried by the air leaving, water can not be employed in the final stage. In such cases, a spray of calcium brine is employed in the final stage. The salt is kept at a constant specific gravity by passing a small portion of it continuously through an evaporator or vacuum pan similar to those used in the manufacture of sugar, an open evaporator being used where live steam is employed for the concentration, and a vacuum pan where exhaust steam is used. By maintaining a constant vacuum and regulating the temperature of boiling thermostatically, the strength of the solution is uniformly maintained.

The dewpoint to which the air is cooled is regulated automatically by a thermostat as described in Par. 18.

The use of an interchanger placed between the stages using refrigerated water or brine and the stage using water from natural sources, makes a material saving in the refrigeration required.

The advantages over the coil or surface type of dehumidifier are: (*a*) reduction in amount of power required; (*b*) reduction of first cost; (*c*) reduction in space required, and the constant and uniform condition of the air supplied to the furnace.

The second subject is that of the dehumidifying of compressed air. The dewpoint for a given weight of water vapor in a given space is

dependent upon the vapor pressure and not upon the air pressure. By compressing the mixture of air and water vapor, the dewpoint required for the same final relative weights of vapor and air will be raised. To illustrate, if it is desired to reduce the water vapor to 2 grains per cu. ft. of free air, and the dehumidifying is done at atmospheric pressure it is necessary to reduce the air to a temperature of approximately 30 deg. fahr. If, however, the air is compressed to double the absolute pressure, for the same results, it will be necessary only to cool to a dewpoint of approximately 50 deg. fahr.

The elevation of the temperature plane not only allows of a high suction pressure on the ammonia compressor with the resultant economy, but it makes it possible to do a very large portion of the work by water from any convenient source, such as rivers, lakes or wells. Of course, the colder the water available, the less refrigeration required.

There are two systems of dehumidifying under pressure which have been designed for blast-furnace application, only one of which, I believe, has been put in operation. In both of these systems, the work is done with the sprays. In one a constant air pressure is maintained in the dehumidifier by a regulating valve placed in the blast pipe leading to the furnace, and a constant dewpoint is regulated by the control described by Mr. Carrier in Par. 18.

The other system provides for varying automatically the dewpoint temperature of the air leaving the dehumidifier, in relation to the variations in the air pressure, to procure a constant relation of the weights of water vapor and air.

The relation between the air pressure and the dewpoint required is a variable, but an automatic regulator has been devised to overcome this difficulty.

With compressed air the efficiency of the interchanger is greatly increased due to the greater density. With an interchanger having the same first cost as the refrigeration which it displaces, the refrigeration required is reduced from 30 to 40 per cent.

With the compressed air method then, there are three factors which reduce the amount of refrigeration and power required: (*a*) the large amount of work that can be done by water from natural sources; (*b*) the high efficiency of an interchanger; (*c*) the high ammonia suction pressure.

The principal advantages of dehumidifying for blast furnaces on the compression side of the blowing engine are: (*a*) a very low first cost; (*b*) small space required; (*c*) low cost of maintenance.

DISCUSSION

THE AUTHORS. A further study of the transmission of heat from fan-coil heaters was made after this paper had been prepared, to ascertain if the theory of transmission therein developed was applicable in case steam at a higher temperature and pressure was used. This second series of tests was made with a steam pressure of 50 lb. in the coils, and the results obtained seem of sufficient interest to warrant a brief summary of a few of the more important points being presented here (Table 13).

The apparatus and method used in the second series were the same as those already described for tests using steam at 5 lb. in the heater, both series of tests having been made at the same time and by the same observers. The theory was also applied to cast-iron heating surface in the form of Vento heaters, the data used being taken from the catalogue of the American Radiator Company. This investigation has been fully described and the results given in a paper by Frank L. Busey[1] on Heat Transmission with Indirect Radiation.

It is noticeable that the value of K for the 50 lb. tests is less than for the tests at 5 lb. steam pressure, which is in accord with the theory that at a higher film temperature the film resistance is greater and the transmission of heat correspondingly less. When these values are reduced to the 625-deg. basis the values of the coefficient of transmission, K_0 in Table 13, become practically identical for the 5 lb. and 50 lb. tests at the corresponding velocities. From this it will be seen that the values to be supplied for the constants in the rational formula [29] are the same for both 5 lb. and 50 lb. pressure, with the pipe coil heaters.

With the Vento heaters, however, the values for K_o are lower and a curve plotted for these conditions, similar to that in Fig. 35, is found to have a slope indicated by

$$\frac{B}{C_p W_o} = 61.00$$

[1] Presented before the January 1912 meeting of the American Society of Heating and Ventilating Engineers.

so that formula [29] becomes

$$K_o = \frac{1}{0.047 + \frac{61.00}{V_o}}$$

where $\theta' = 625$ deg. absolute.

The results from the Vento tests show a higher film temperature and resistance and a lower rate of transmission than with the pipe coil heaters. The frictional resistance is also less and it is probable that the rate of transmission varies as the frictional resistance to the pas-

TABLE 13 COMPARISON OF THE VALUES OF θ'_m, R, K AND K_o

Buffalo Standard W. I. Pipe Heaters									Vento Regular C. I. Heaters 5 in. Centers				
Steam Pressure, 5 Lb.					Steam Pressure, 50 Lb.				Steam Pressure, 5 Lb.				
Velocity	Mean Absolute Film Temperature, θ'_m	Film Resistance, R	Coefficient of Transmission, K	Coefficient of Transmission Corrected to Standard Film Temperature of 625 Deg. Abs., K_o	Mean Absolute Film Temperature, θ'_m	Film Resistance, R	Coefficient of Transmission, K	Coefficient of Transmission Corrected to Standard Film Temperature of 625 Deg. Abs., K_o	Velocity	Mean Absolute Film Temperature, θ'_m	Film Resistance, R	Coefficient of Transmission, K	Coefficient of Transmission Corrected to Standard Film Temperature of 625 Deg. Abs., K_o
200	669.9	0.0479	3.316	3.354	733.5	0.0525	3.322	3.413	200	672.6	0.0511	2.795	2.819
300	662.4	0.0474	4.674	4.735	723.3	0.0517	4.549	4.706	300	666.1	0.0507	3.948	3.990
400	655.7	0.0469	5.788	5.858	714.0	0.0510	5.626	5.838	400	660.4	0.0503	4.917	4.990
500	649.4	0.0465	6.743	6.841	705.7	0.0505	6.565	6.839	500	654.8	0.0497	5.837	5.922
600	643.2	0.0460	7.614	7.718	697.8	0.0449	7.461	7.784	600	649.8	0.0494	6.738	6.825
700	638.3	0.0456	8.412	8.476	691.5	0.0495	8.064	8.403	800	640.6	0.04875	8.051	8.145
800	633.5	0.0454	9.146	9.204	684.2	0.0489	8.829	9.190	1000	632.6	0.0480	9.191	9.255
900	629.0	0.0450	9.823	9.850	678.4	0.0485	9.472	9.846	1200	626.0	0.0476	10.160	10.200
1000	624.7	0.0447	10.465	10.455	672.1	0.0481	10.063	10.440	1400	619.8	0.0471	11.030	11.050
1100	620.1	0.0444	11.040	11.000	666.7	0.0477	10.640	11.020	1600	614.3	0.0467	11.780	11.700
1200	616.3	0.0441	11.610	11.490	.....	.....	.....	.....	1800	609.0	0.0463	12.505	12.350
Total Variation	8.00%	7.93%	.....	.....	9.17%	9.14%	.....	.....	.....	9.46%	9.39%	.....	.....

sage of air through the clear area of the heater. Through the ordinary working range the Vento heaters require a velocity approximately 24 per cent greater than the pipe heaters.

The following general conclusions may be deduced:

a The authors' theory of heat convection appears to be established from four separate and independent sources. Apparently there is a true film resistance to the transmission of heat which varies inversely as the density of the

air or gas in the surface film and hence directly as the absolute film temperature.

b It has been conclusively shown that the rate of heat transmission per degree difference in temperature between the steam and air is not constant for a given velocity, but varies in accordance with the theoretical film temperature.

c This effect of film resistance has a very important bearing in many other cases of heat transmission, as for example, in the indirect transmission of heat in boilers and in the cooling of gases in the cylinders of gas engines. In such cases there are very high film temperatures comparatively, which tend to greatly reduce the rate of transmission. At these high film temperatures, for the same reason, the per cent increase in the rate of transmission with increased velocities is greatly reduced.

d The film resistance is the same for all surfaces regardless of their form, but that portion of the transmission resistance which is dependent on the velocity is inversely proportional to the effective velocity and will not be the same for any two designs of heaters.

e In general it may be stated that when the velocities in the different types of heaters are so modified as to obtain the same frictional resistance, the transmission resistance will be the same. For example, Vento heaters are shown to give a considerably lower transmission than staggered pipe coils at the same air velocity, owing to the difference in the disposition of the surface. But for the same reason the Vento also has considerably lower frictional resistance, so that by increasing the velocity to give the same frictional resistance, the rate of transmission becomes substantially the same in both styles of heaters.

No. 1342

SOME EXPERIENCES WITH THE PITOT TUBE ON HIGH AND LOW AIR VELOCITIES

By Frank H. Kneeland, Gary, W. Va.

Junior Member of the Society

As is well known the Pitot tube is an instrument for measuring the velocity of moving fluids. Its general principle is well understood; but we believe that, possibly on account of fancied difficulties in its manipulation, it has never received thoughtful consideration from the majority of engineers, nor been put to the practical use to which its merits justly entitle it. This paper, therefore, will deal rather more with the practical side of the instrument than with the theoretical.

2 The tests herein described were made in part in connection with the flow of air through an experimental pipe line erected for the study of the problem of conveying coal by means of an air blast. The idea of transporting coal by this means from the workings of the mine to a tipple or bin is not a new one, several patents having been issued for such schemes. During the spring of 1909 one of the officials of the United States Coal & Coke Company conceived the idea of transporting the coal cut by a mining machine, by means of suction, on a principle similar to the Darley ash conveyor, or the better known vacuum carpet sweeper. Under certain conditions, this method of mining possesses many obvious advantages over present practice.

3 After carefully studying this problem in the light of available data, it appeared so good on paper that it was deemed justifiable to build an experimental plant for the purpose of determining:

a The velocity of air current necessary to carry coal in pipes of a given diameter.

b The amount of coal which could be transported by a given amount of air, in pipes of given diameter, but at varying velocities.

Presented at the Annual Meeting 1911, of The American Society of Mechanical Engineers.

c The loss of pressure due to friction of air in the pipes while carrying coal.

4 Accordingly the plant shown in plan and profile in Fig. 1 was constructed. It consisted essentially of a mining machine, a mine pipe for transporting the coal, a suction head to separate the coal from the air and a positive blower, or rather a "sucker," for creating the air blast. The mining machine above referred to cut the coal in sizes ranging from fine powder to lumps equivalent to about 2-in. cubes. The machine used in our experiments was originally designed for loading the coal into mine cars and was, therefore, equipped with a conveyor extending from its rear end. In our early experiments, it discharged its coal into a hopper mounted on 6-in., 8-in. or 12-in. ordinary wrought-iron pipe, as shown in Fig. 2. Later on it was arranged to discharge into a pipe of rectangular section, with approx-

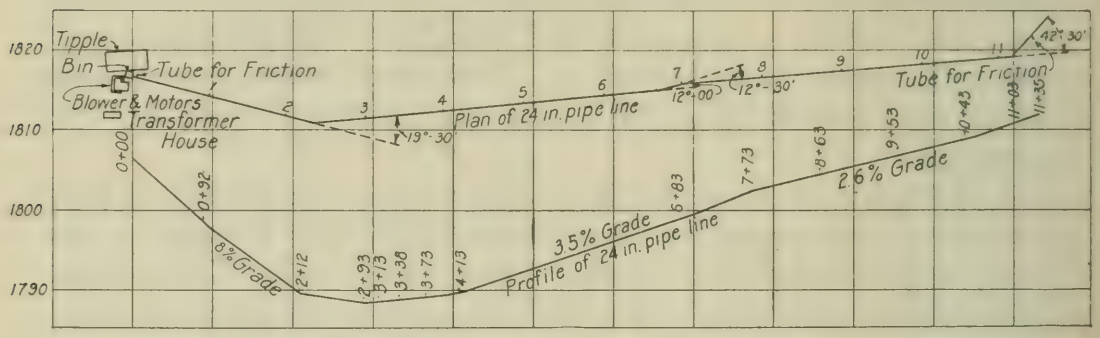


FIG. 1 PLAN AND PROFILE OF EXPERIMENTAL PIPE LINES

imately the same area as the 24-in. mine pipe. As will be seen in Fig. 2, above mentioned, only two pipes were used at a time, one carrying the coal and the other for the purpose of "spilling in" air to the 24-in. main mine pipe, to which the two smaller pipes were connected by means of a plate flange. The 24-in. mine pipe was made up of 30-ft. sections, except sections of special length, of which three were used. Each section was composed of six plates, each plate extending throughout the complete circumference and being lapped and riveted at the joint. Each of these longitudinal seams were placed 180 deg. from those of the plates next joining it. All transverse or girth seams were single riveted at the joints; all edges of plates inside the pipe were beveled and all rivets countersunk on the inside. Each section or length of pipe was provided with wrought-iron flanges at each end riveted to the pipe, the flange rivets being countersunk on the inside like all the others. The sections were put together with $\frac{1}{8}$-in. rubber gaskets between all flanges, which made an extremely tight mine

pipe and one on which repeated tests as the work progressed failed to show any perceptible leakage.

5 The outer or tipple end of this mine pipe was connected with a square wooden suction head. At the upper end of the suction head a 36-in. cast-iron pipe connected it to a No. 11 Roots positive blower. This machine was driven through gearing by two 200-h.p. 720-r.p.m. induction motors. Two ratios of gearing were employed, one driving the blower somewhat below 90 r.p.m. and the other somewhat above 100 r.p.m. The normal displacement of the blower was 300 cu. ft. per revolution. This, however, was when the machine was operating under no pressure. Fig. 3 shows its capacity in cubic feet of free air per revolution under varying vacua. In the 36-in. pipe line and directly above the blower was a 36 in. by 16 in. by 36 in. tee with its 16-in. branch looking up. On the upper flange of this tee was

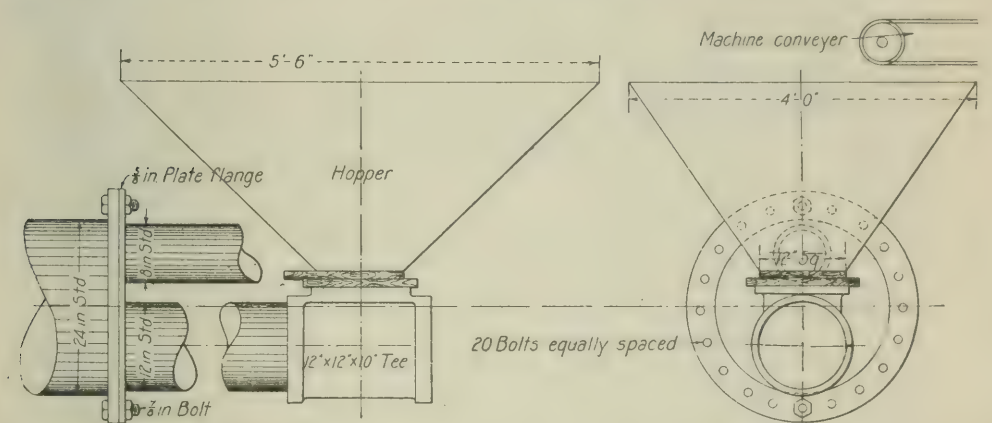


FIG. 2 ARRANGEMENT OF 8-IN. AND 12-IN. PIPES AND HOPPER AT END OF 24-IN. PIPE LINE

mounted a 16-in. straightway valve, upon which was placed a length of 16-in. pipe somewhat over 12 ft. long. The whole external arrangement except the motors and gearing may be clearly seen in Fig. 4.

6 The experimental plant has been carefully described in order that the manipulation of apparatus, etc. may be clearly understood. The velocity of air in a pipe carrying coal or other foreign substances cannot be measured by any direct means, and we, therefore, had to resort to indirect; that is to say, to determine the capacity of the blower per revolution and deduct from the quantity handled by the blower, the sum of the leakage and admission of air other than that through the mine pipe carrying the coal. The resulting quantity was, of course, that passing through the coal-carrying pipe. There were four points or places of admission of air, namely, the coal pipe,

the admission pipe parallel to it, the admission above the blower, and the suction head. Although this suction head was built up of a double thickness of tongued and grooved flooring with all joints laid in white lead, it leaked considerably and it became necessary before the experiments were over to cover the whole structure completely with tar paper laid in thick tar paint. Tests for leakage in this part of the apparatus were run at frequent intervals and corrections made for it in all calculations. It was necessary then while using the two pipes at the end of the mine pipe, to measure the velocity of the air at two places, in one of the pipes above mentioned and in the pipe above the blower. In order that all observations might be taken simultaneously, a system of signals was installed whereby the closing of a small knife switch at a point near the mining machine flashed

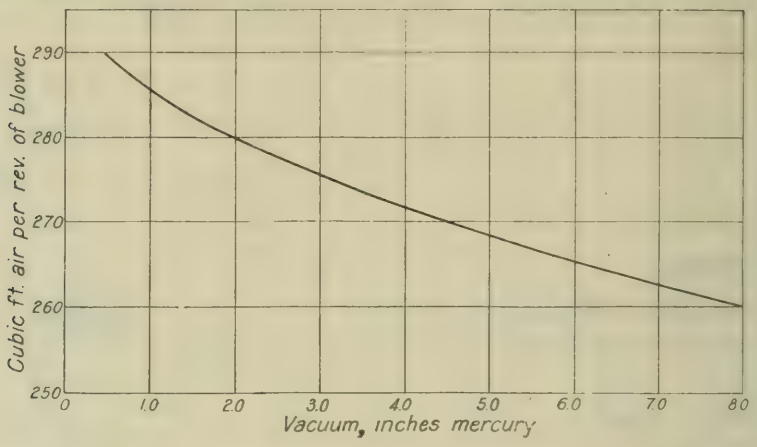


FIG. 3 CURVE FOR DISCHARGE OF ROOTS BLOWER PER REVOLUTION AT VARIOUS VACUA

ordinary incandescent lamps in front of each observer, both inside and outside the mine. Not only were observations taken on the velocity of air but also at three different points on the vacuum as well. A direct-reading water manometer was connected to the 24-in. pipe a short distance behind the mining machine; a differential mercury manometer, multiplying five times, was attached to the 24-in. pipe just outside its connection with the suction head, as shown at *A* in Fig. 4, and an ordinary direct-reading U-tube manometer attached to the 36-in. pipe a short distance above the blower, as shown at *B* in Fig. 4. In addition to the manometers above mentioned, there were used in connection with the Pitot tubes glass manometers of the U-type where one leg is enclosed within the other. These instruments were 16 in. long and provided with scales reading to one-tenth of an inch. Either water or mercury was used as a working fluid, depend-

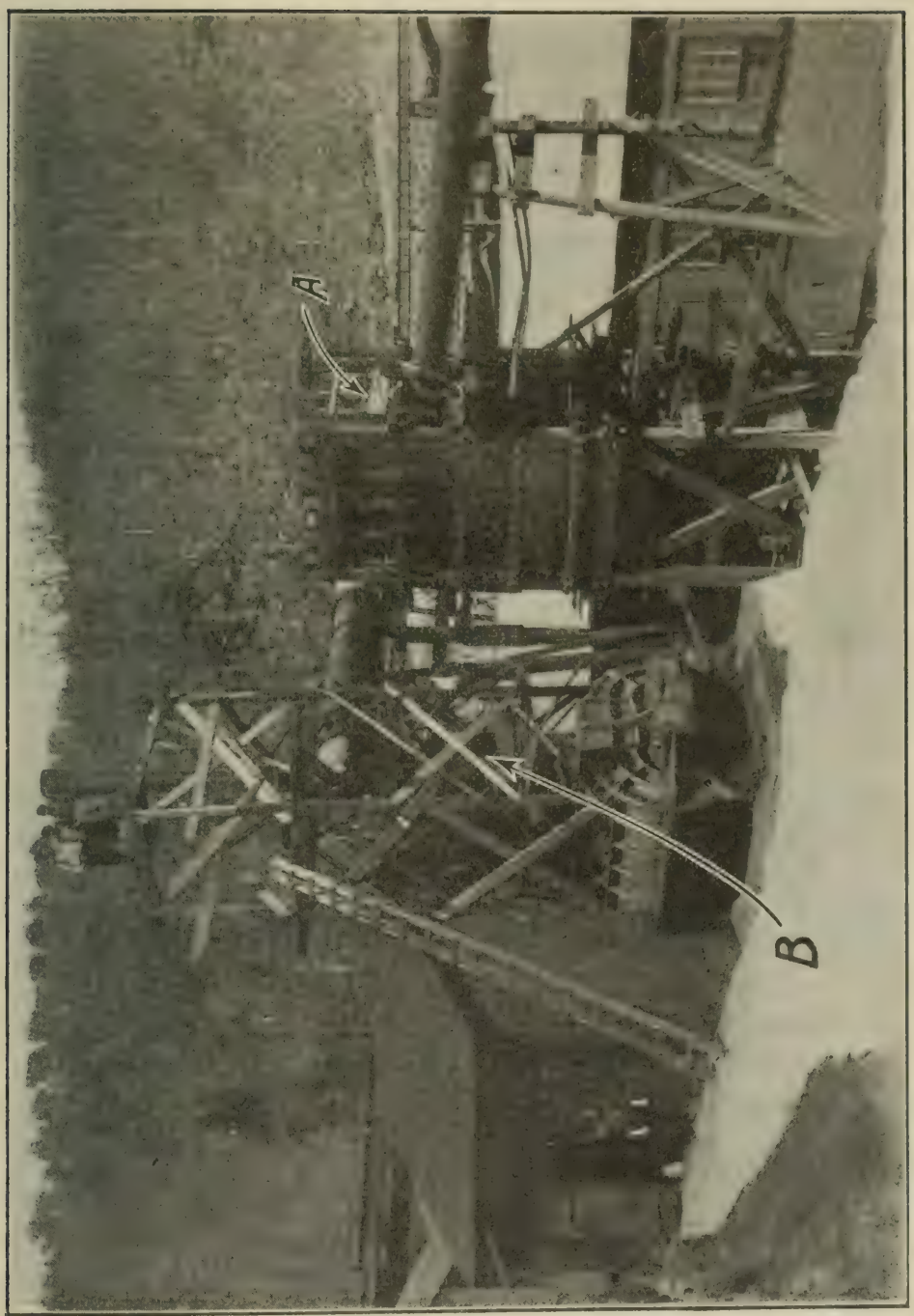


FIG. 4 GENERAL VIEW SHOWING BLOWER AND CONNECTIONS WITH PIPE LINES

ing upon the velocity of the air being measured. Although not appearing on the data sheets, corrections were always made for capillarity of the fluid in the manometers.

ON THE USE OF THE PITOT TUBE

7 In general the Pitot tube consists essentially of two devices: One (*a*) for converting the energy of a moving fluid into a pressure; and the other (*b*) for measuring the pressure on the fluid under consideration. In this paper the first will be known as the Dynamic Nozzle and the pressure registered by it will be called the Dynamic Head; and the second as the Static or Piezometer Nozzle and the pressure registered by it will be called the Static Head. In our work at Gary, we used six tubes of three distinct types, shown in Fig. 5, known as the United States No. 1, the Gebhardt Nos. 1 and 2, and the Taylor Nos. 1, 2 and 3. The United States No. 1 tube was made at our own shops and consisted of the dynamic nozzle *A* and the static nozzle *B*. In use the dynamic nozzle was placed in the holder *C*, which was screwed firmly into the wall of the pipe in which a test was being made. It was clamped into the holder by the small set screw shown in the drawing (Fig. 5). The static nozzle was screwed into the wall of the pipe, like the holder above mentioned, until its inner end came flush with the inner surface of the pipe. The static and dynamic nozzles of this tube were always placed on the same girth line of the pipe and approximately 90 deg. from each other. The Gebhardt tubes, designed by Prof. G. D. Gebhardt and purchased from the Armour Institute of Technology, were practically identical with each other. They had been carefully calibrated and re-calibrated in gas mains and were said by their designer to possess a coefficient of unity. It will be observed that the static nozzles of these tubes were beveled; this was done in order to avoid aspiration in this part of the tube. In a paper[1] entitled The Pitot Tube as a Steam Meter, when speaking of the beveling of the static nozzle of the tube to prevent aspiration, Professor Gebhardt stated that further experiments were necessary to show whether any fixed angle is applicable to all velocities. Our experiments would bear out the above statement as this seems to be the weak or inaccurate part of the apparatus. The other details of the tube are shown with sufficient clearness in the drawing.

8 The Taylor tubes were, like the Gebhardt tubes, practically

[1] Trans. Am.Soc.M.E., vol. 31, p. 601.

identical with each other. These tubes were kindly loaned to us by Capt. D. W. Taylor, naval constructor at the United States Navy Yard, Washington, D. C., and had been used very successfully in testing ventilating fans for warships. This type of tube was very much more difficult to manipulate than the other two. Due to its shape, it had to be adjusted with extreme care when dealing with the high velocities employed by us, and at such times the writer and his associates were in constant fear lest its long point, or nose,

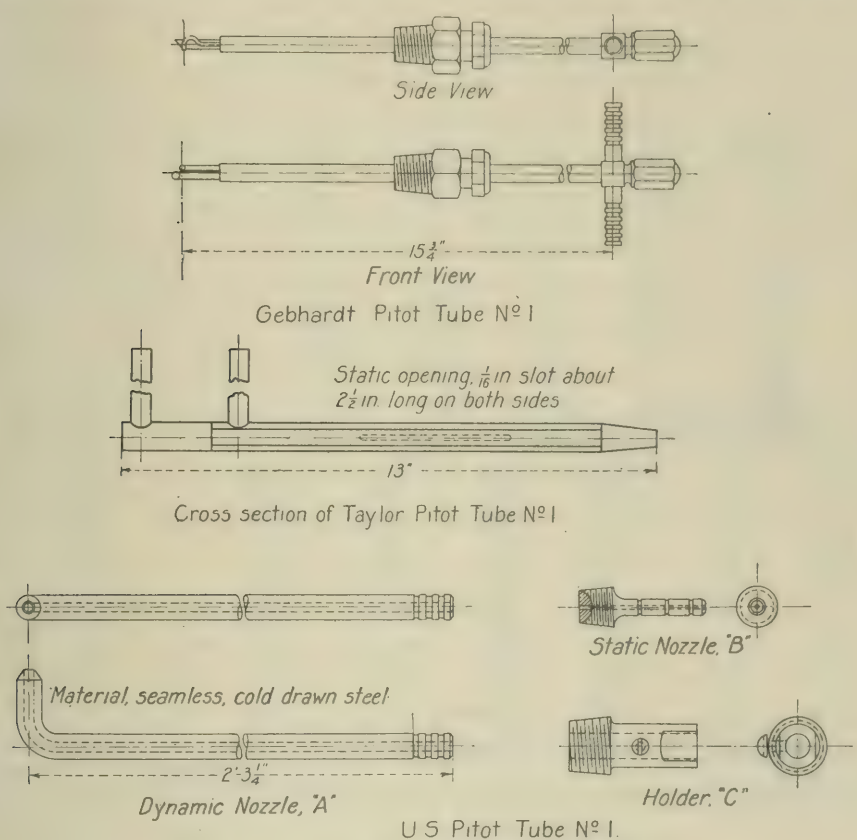


FIG. 5 TYPES OF PITOT TUBES USED

be turned sidewise and the tube capsize, so to speak. Fortunately no such accident occurred, but it is possible that some of the erroneous readings of these tubes may have been caused by the tube taking a slightly sidewise position when subjected to the enormous wind pressures developed. It might, also, be mentioned that we made three other tubes, none of which were ever used on account of their large size and consequent clumsiness and difficulty of manipulation. Also in one test five dynamic nozzles were used, similar to those of the United States No. 1 tube, arranged one in each of the five positions shown in Table 1.

9 These five nozzles were connected to the automatic averaging block shown in Fig. 6, which in turn was connected to a manometer. The static pressure was obtained from a static nozzle exactly similar to that of the United States No. 1 tube. This arrangement of the apparatus, although apparently possessing some advantages over the ordinary method, is not only rather clumsy but is also inherently inaccurate, since the average velocity in a pipe is not dependent on the square root of the average dynamic pressures obtained at any

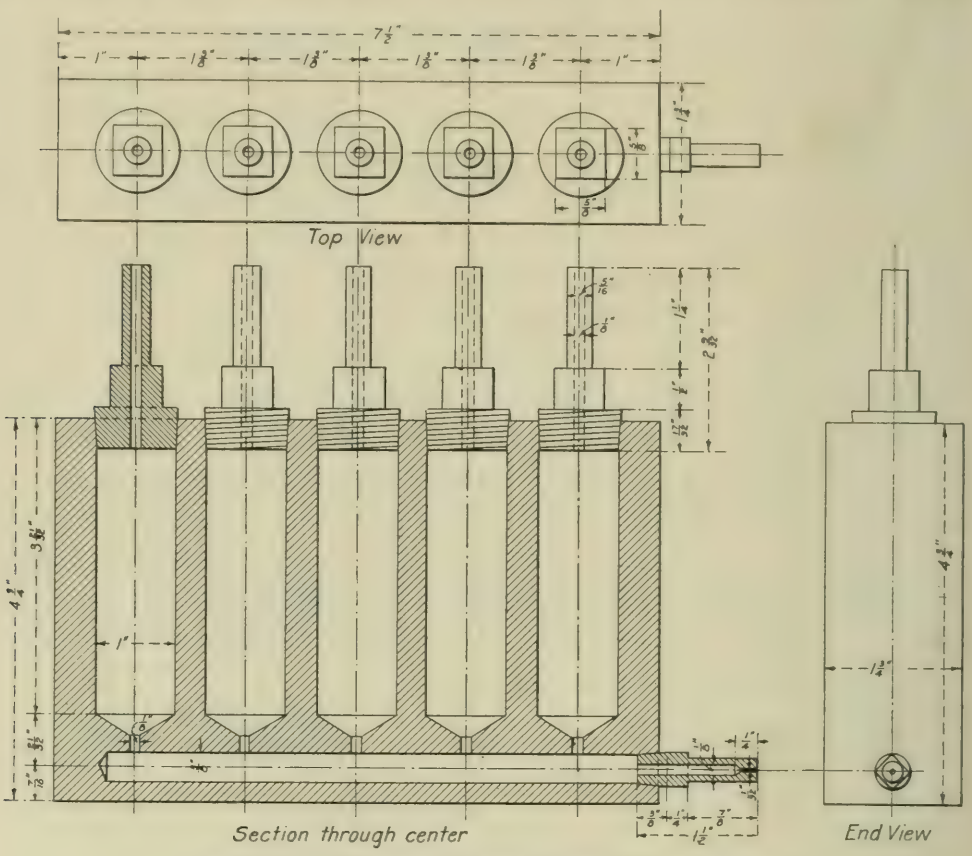


FIG. 6 AUTOMATIC AVERAGING MANOMETER

given number of points, but upon the average square roots of these pressures. It will also be seen that the liability of errors caused by leakages in the joints between the instrument and rubber tubing connecting it to its averaging block and manometer is at least six times as great in this apparatus as in the simpler tubes.

11 To obtain the average velocity existing in a pipe from the use of a Pitot tube, recourse may be had to two methods. The first of these is to divide the diameter of the pipe into a given number of equal parts and take one or more readings at each division point. Thus, supposing the work is being done with a 16-in. pipe, the tube

would first be placed as near the side wall of the pipe as possible and a reading, or a series of readings, taken. It would then be placed, say 1 in. from the side wall and an equal series of readings taken, and so on across the diameter of the pipe. The average velocity would then be considered as the average of the different velocities obtained at the several points of observation. It will readily be seen that this average is not strictly true, since the outer rings of the cross-section are of equal width with the inner ones, consequently, they are unequal in area, yet all were given equal weight in the calculations. The second, and better method, is to divide the cross-section of the pipe into a given number of equal areas and place the dynamic nozzle of the tube in the center of gravity, so to speak, of the circle being tested. To again illustrate: The pipe used above the blower in our experiments had an external diameter of 16 in., the wall being $\frac{1}{4}$ in. thick. This gave an internal diameter of $15\frac{1}{2}$ in. and an area of 188.69 sq. in. Dividing this into five equal parts, each

TABLE 1 PIPE SIZES AND POSITIONS OF TUBE

Nominal Pipe Size, In.	Actual Internal Diameter, In.	Internal Area		First Position	Second Position from Center, In.	Third Position from Center, In.	Fourth Position from Center, In.	Fifth Position from Center, In.
		Sq. In.	Sq. Ft.					
8	$8\frac{3}{8}$	55.088	0.382	Center	$2\frac{1}{4}$	$2\frac{15}{16}$	$3\frac{1}{2}$	$3\frac{15}{16}$
12	12	107.33	0.745	Center	$3\frac{5}{16}$	$4\frac{1}{8}$	$4\frac{7}{8}$	$5\frac{1}{2}$
16	$15\frac{1}{2}$	188.69	1.31	Center	$4\frac{3}{16}$	$5\frac{7}{16}$	$6\frac{7}{16}$	$7\frac{5}{16}$

part would have an area of 37.738 sq. in. Taking the first position of the Pitot tube at the center of the pipe, we assume that the velocity of air is constant over the circle or disc whose area is 37.738 sq. in. The second position of the tube is found by dividing the next ring or part into two equal areas and adding one of these to the central disk, and finding the radius of this sum. In the above case expressed mathematically this becomes

$$R^2 = \frac{37.738 + \frac{37.738}{2}}{3.1416}$$

$$R = \sqrt{\frac{37.738 + \frac{37.738}{2}}{3.1416}}$$

$$= 4\tfrac{3}{16}$$

12 These positions for the different sized pipes used, as well as the principal dimensions of the pipes, are shown in Table 1.

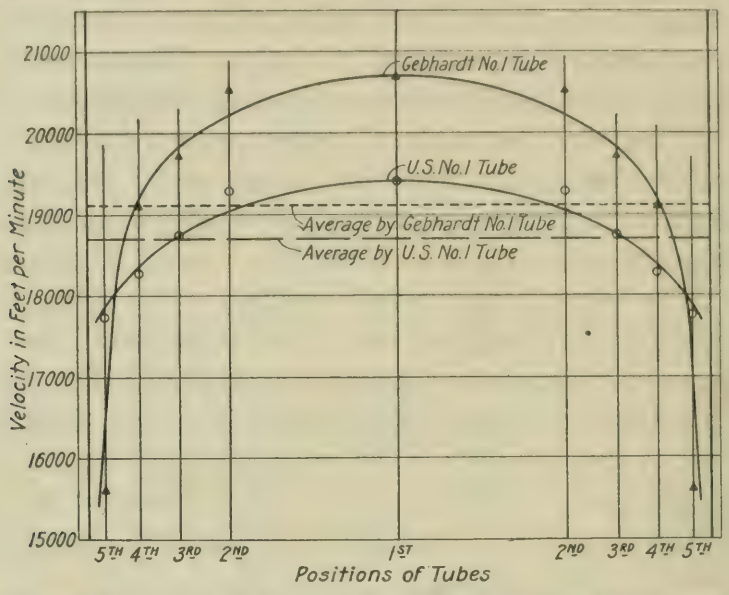


FIG. 7 CURVES SHOWING VELOCITY CHANGES ACROSS DIAMETER OF PIPE WITH GEBHARDT NO. 1 AND UNITED STATES NO. 1 PITOT TUBES

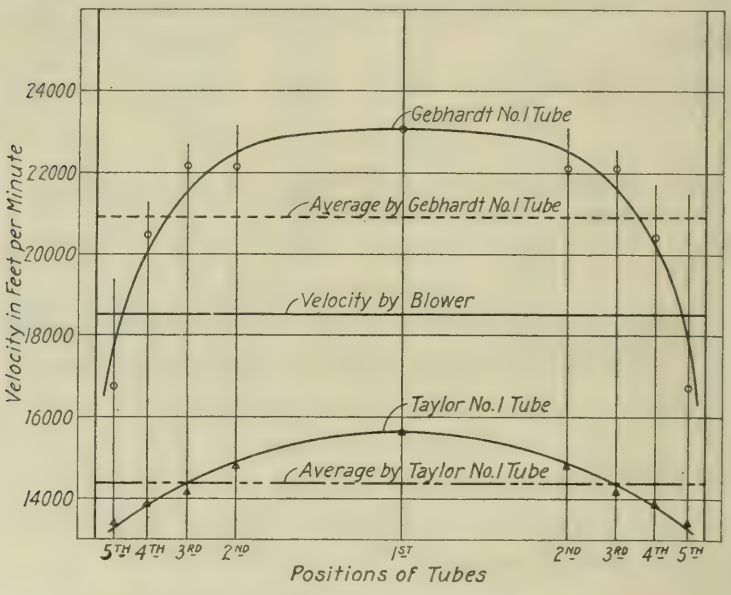


FIG. 8 SHOWING VELOCITY CHANGES ACROSS DIAMETER OF PIPE WITH GEBHARDT NO. 1 AND TAYLOR NO. 1 PITOT TUBES; ALSO VELOCITY AS SHOWN BY BLOWER

13 In Figs. 7, 8 and 9 are plotted the results of tests in the 16-in. pipe above the blower showing how the velocities vary across the diameter of the pipe as determined by the different tubes used. The data sheets from which these curves are plotted are found in the ap-

pendix. These are typical and fair averages of all tests made on the different sized pipes used.

14 The common formula for the Pitot tube is

$$v = \sqrt{2gh}$$

where

v = velocity in ft. per sec.
g = acceleration due to gravity in ft. per sec.
h = head in ft. causing flow

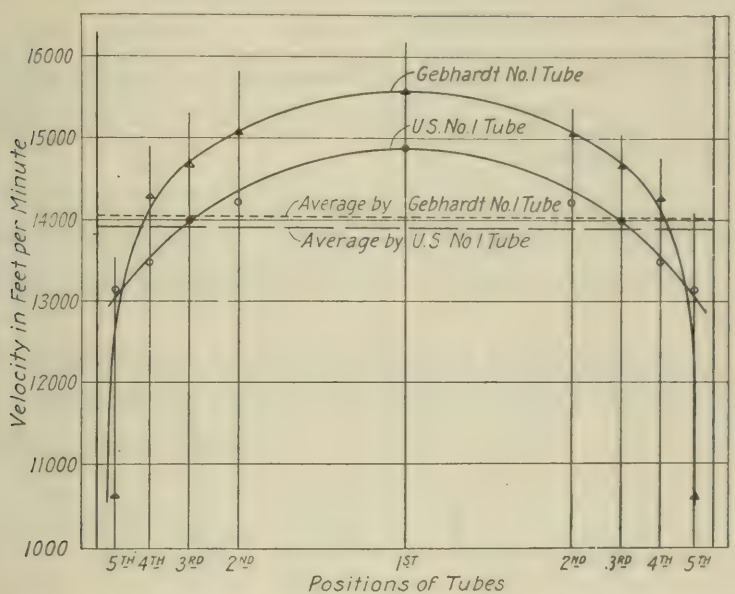


FIG. 9 CURVES SHOWING VELOCITY CHANGES ACROSS DIAMETER OF PIPE WITH GEBHARDT NO. 1 AND UNITED STATES NO. 1 PITOT TUBES

But in air measurement h is a head of air in feet but is measured by h_1, which is a head of water (or mercury) in inches. Denoting the weight of the air per cubic feet by w, taking the density of water as 62.4 lb. per cu. ft. and letting V = velocity of flow in ft. per min. we have

$$V = 60\sqrt{2gh}$$

$$= 60\sqrt{\frac{2 \times 32.16 \times 62.4 \times h_1}{12 \times w}}$$

$$= k\sqrt{h_1}$$

Now

$$w = \frac{1.3253 \times B}{459.2 + T}$$

where

B = height of barometer

T = temperature, deg. fahr.

1.3253 = weight in lb. of 459.2 cu. ft. of air at 0 deg. fahr. and 1 in. barometric pressure.[1] This gives the weight of dry air.

15 In the Smithsonian Meteorological Tables[2] a formula is given for the weight of humid air which, changing metric to English units, is

$$P = \frac{0.080723}{1 + 0.0020389\ (t - 32)} \times \frac{b - 0.378\,e}{29.921}$$

where t is temperature in deg. fahr., b is corrected height of barometer in inches of mercury, and e is the pressure due to the vapor in the air in inches of mercury.

16 This formula was used for determining the weight of air in all of our calculations; the humidity of the air being determined by psychrometric observations and the use of the Smithsonian tables above referred to.

17 In a paper entitled Experiments with Ventilating Fans and Pipes[3] Capt. D. W. Taylor, naval constructor, gives an exact formula for the Pitot tube. The following is quoted in part from Captain Taylor

$$\frac{v_1^2 - v_2^2}{2g} = \frac{y}{y-1} \times \frac{p_2}{P_2}\left\{1 - \left(\frac{p_1}{p_2}\right)^{\frac{y-1}{y}}\right\}$$

where

v_1 = velocity in ft. per sec. at a point where the pressure = p_1 in lb. per sq. ft., p_2 = pressure in lb. per sq. ft. at any other point

v_2 = velocity

and

P_2 = weight in lb. per cu. ft. where pressure = p_2

y = ratio between specific heats of air under constant pressure and constant volume = 1.408

g = acceleration due to gravity in ft. per sec.

18 Now if p_2 is the pressure at the impact opening of the Pitot tube, $v_2 = 0$. Also take p_1 as the pressure in the unchecked stream measured by the manometer connected with the pressure openings on the side of the Pitot tube. Also let p_a, P_a, T_a be the pressure,

[1] Kent's Mechanical Engineers Pocket Book, ed. 7, p. 481.

[2] p. 55.

[3] Society of Naval Arch. and Marine Engrs., 1905, p. 35.

weight per cubic foot and absolute temperature of the atmosphere; T_1 and T_2 absolute temperatures corresponding to p_1 and p_2. Then

$$\frac{p_2}{P_2T_2} = \frac{p_1}{P_1T_1} = \frac{p_a}{P_aT_a} = x$$

and from well-known formulae for air

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{y-1}{y}}$$

the change of pressure from p_1 to p_2 being adiabatic. Then from the above

$$\frac{p_2}{P_2} = xT_2 = xT_1\left(\frac{p_2}{p_1}\right)^{\frac{y-1}{y}}$$

$$\frac{{v_1}^2}{2g} = \frac{y}{y-1}\,xT_1\left\{\left(\frac{p_2}{p_1}\right)^{\frac{y-1}{y}} - 1\right\}$$

Let

$$p_2 = p_1(1+k)$$

Then

$$\frac{{v_1}^2}{2g} = \frac{y}{y-1}\,xT_1\left\{(1+k)^{\frac{y-1}{y}} - 1\right\}$$

Expanding and reducing

$$\frac{{v_1}^2}{2g} = xT_1\left\{k - \frac{k^2}{2y} + k^3\,\frac{1+y}{6y^2} - k^4\,\frac{(1+y)\ (1+2y)}{24y^3}\right\}$$

Now

$$y = 1.408$$

$$\frac{1}{2y} = 0.355$$

Therefore

$$v_1 = \sqrt{2g\ \frac{p_1k}{P_1}\left\{1 - \frac{k}{2y} + k^2\,\frac{1+y}{6y^2} - k^3\,\frac{(1+y)\ (1+2y)}{24y^3}\right\}}$$

19 Let V_1 = velocity in ft. per min. and the readings of p_2 and p_1 be expressed in in. of mercury instead of in lb. per sq. ft. Then

$$V_1 = 60\sqrt{64.32 \times \frac{p_2-p_1}{P_1}\left\{1 - \frac{k}{2y} + k^2\,\frac{1+y}{6y^2} - k^3\,\frac{(1+y)\ (1+2y)}{24y^3}\right\}} \times 70.704$$

$$= 4046.16\sqrt{\frac{p_2-p_1}{P_1}\left\{1 - \frac{k}{2y} + k^2\,\frac{1+y}{6y^2} - k^3\,\frac{(1+y)\ (1+2y)}{24y^3}\right\}}$$

$$= 4046.16\sqrt{\frac{p_2-p_1}{P_1}\left\{1 - 0.355\,k + 0.202\,k^2 - 0.137\,k^3\right\}}$$

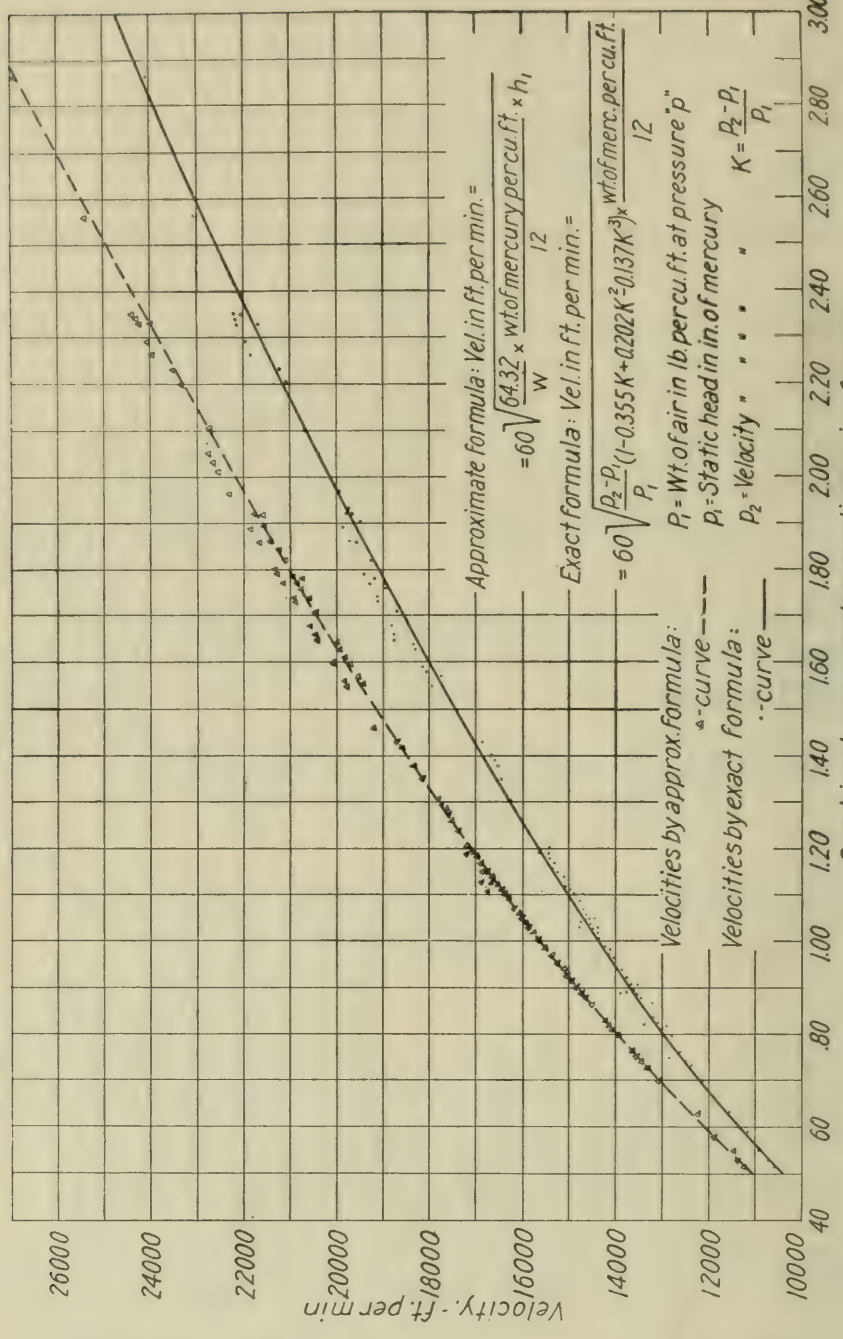


FIG. 10 CURVES FOR AIR VELOCITIES AS CALCULATED BY APPROXIMATE AND EXACT PITOT TUBE FORMULAE

The values above given are for a temperature of 70 deg. fahr., a barometer of 29.921 in., and a humidity of 70 per cent.

20 Curves in Fig. 10 show a comparison of the velocities of air as calculated by the two formulae already given. It will be seen that at a velocity of 23,000 ft. per min. these formulae give results varying by over 15 per cent, while at 11,000 ft. per min. the variation is only about 6 per cent. For velocities below 6000 ft. per min. there seems to be no good reason for using the exact formula as given above, since the difference between it and the common formula is well within the limit of error in observation.

FRICTION IN EXPERIMENTAL PIPE LINE

21 When designing the experimental plant, calculations were

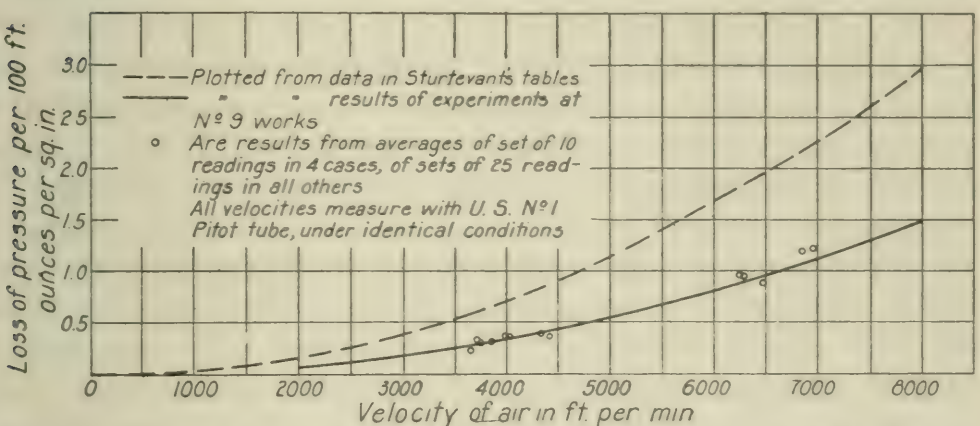


FIG. 11 CURVES OF LOSS OF PRESSURE DUE TO FRICTION IN 24-IN. PIPE

made for the loss of head due to friction in the pipe line. These calculations were based on Sturtevant's tables on air friction, since they were the most reliable data at hand. The experiments on the plant, however, gave an excellent opportunity to test not only the loss due to friction when carrying varying amounts of coal, but to check the calculated loss when the air was flowing alone and unburdened. Accordingly, observations were taken on the manometers at each end of the 24-in. mine pipe with this end in view, the results of which are shown in Fig. 11. This, however, does not give the loss from friction when carrying coal.

22 Some time after conducting the experiments described on the flow of air in pipes, the necessity arose for accurate measurement of the flow of air in mine headings in connection with fan tests, also of the air discharged from the chimney or outlet of fans, and we naturally turned to the Pitot tube as being the most accurate means at hand.

USE OF PITOT TUBE IN MINES

23 When we come to use the Pitot tube in mines we find several conditions not existing in large or small pipes. The floor, ribs and roof of a heading or room are always more or less rough and uneven, which causes innumerable eddies and swirls in the air current similar to those seen in the water along the sides of a swift stream flowing over a rocky or uneven bed. It is impossible, also, to determine from the appearance of an airway just where or why you will find the stronger air current. The velocity at which the air is flowing may also cause a decided difference in the contour of average velocity even though all other conditions remain the same. This is well illustrated in fan tests Nos. 9 and 10, Figs. 13 and 14 in the appendix. Of course, the ideal section of a heading in which to take a test would be a long, straight, smooth, rectangular passage. This can never be found and the nearest approximation available is the one to be chosen. At one side of the heading and preferably just behind the test section a "cubby-hole" or operating room should be dug out of the rib, principally for the convenience and comfort of the operator or operators. At the same time, however, the cubby-hole not only shields the operator and makes a convenient place to set up manometers, but also when it is used there are no corrections to be made for the operator's body and no eddy currents are caused by him. By stretching a series of strings or wires from floor to roof and from rib to rib, the test section may be divided into a number of smaller sections of equal, or nearly equal, area. The number of these smaller sections will depend entirely on the area of the test section. It is advisable, however, to have the small sections not much over 4 sq. ft. in area and less if possible. The smaller the sections the greater is the accuracy.

24 For holding the Pitot tube we built a telescoping stand of pipe and pipe fittings with a holding block that could be raised and lowered or clamped rigidly upon either part of the standard. We also made an anemometer holder provided with a spring which held the anemometer out of gear until a string attached to the operating lever was pulled. Upon release of the tension on this string, the spring pulled the instrument out of gear again and stopped its registration. The Pitot tube passed through the holding block and extended a very short distance beyond the anemometer. It will thus be seen that the Pitot tube could be successively placed in the center of each of the small sections and that in each one the anemometer was immediately beside it and subjected as nearly as possible to the same ve-

locity of air as the Pitot tube. The static head of the Pitot tube readings was taken direct on the manometer connected to it, since the whole apparatus was inside the heading in which measurements were being taken and, therefore, subject to the same pressure. Trial was made with the Gebhardt No. 1 tube (in place of the United States No. 1 regularly used) to ascertain if the taking of the static head beside the velocity head or in the cubby-hole on the manometer made any difference in the manometer reading, with the result that the readings in both cases were the same, as might have been expected. We also checked the accuracy of both instruments with smoke and carbon di-sulphide. The method of making this test was as follows:

25 A distance of 200 ft. was measured off along one of the main haulage roads of a mine. The cross-section of this heading was as nearly uniform as a mine heading in this kind of coal (Pocahontas) can well be and the floor and roof were free of obstructions and very even. Midway of this distance and opposite a safety pocket, a hole excavated in the rib for the miners to stand in when a locomotive or trip is passing, the stand bearing the Pitot tube and anemometer was set up and a series of readings taken on both instruments. One observer then stationed himself at the upper end of the 200-ft. distance and another at the lower end. The former was provided with a revolver and blank cartridges loaded with black powder while the latter held a watch. At a signal from the man at the lower end, the one at the upper end of the trial distance fired across the heading and the time was noted when the odor of the powder reached the lower end. The smoke was entirely dissipated before it had traveled half the distance. As a check on the above method, it was repeated, except that instead of using the revolver, a bottle of carbon di-sulphide (CS_2) was used. This was not as satisfactory as the former, since an appreciable amount of time was lost in uncorking the bottle and throwing the liquid into the air. However, the results checked fairly close if an allowance be made for lost time. The above was repeated in a heading with very much lower velocity, about 200 ft., and the Pitot tube found to be almost worthless. The anemometer, on the other hand, gave results which were very fairly accurate.

MEASURING DISCHARGE FROM FAN CHIMNEYS

26 In measuring the discharge from fan chimneys we employed a method very similar to that used in mine headings. Instead of the

telescoping stand, however, we used two $\frac{3}{8}$-in. pipes extending transversely across the fan chimney and held firmly at each end with a specially forged clamp. Upon these pipes or guides was mounted a runner or slide arranged to hold both the Pitot tube and anemometer holder. The top of the chimney was divided into a given number of approximately equal areas by stretching strings or wires across in both directions, as shown in Fig. 12, and the two instruments placed successively in each of the areas. It will be observed from the results of these tests shown in the appendix, that the velocity in these chimneys is always greatest on the tangent side, decreases to a minimum near the center and then increases on the front side. Transversely, the velocity of air seems to follow no fixed law, but in general seems to be greater in the outside areas. When the fan is delivering a large quantity of air also, the velocity near the center of the chimney becomes so low that it cannot be measured with the Pitot tube and is gusty and variable in its nature, the anemometer frequently halting, and even running backward for some seconds at a time. It should be stated also, that all the tests we have made thus far are upon fans of the Clifford-Capel type.

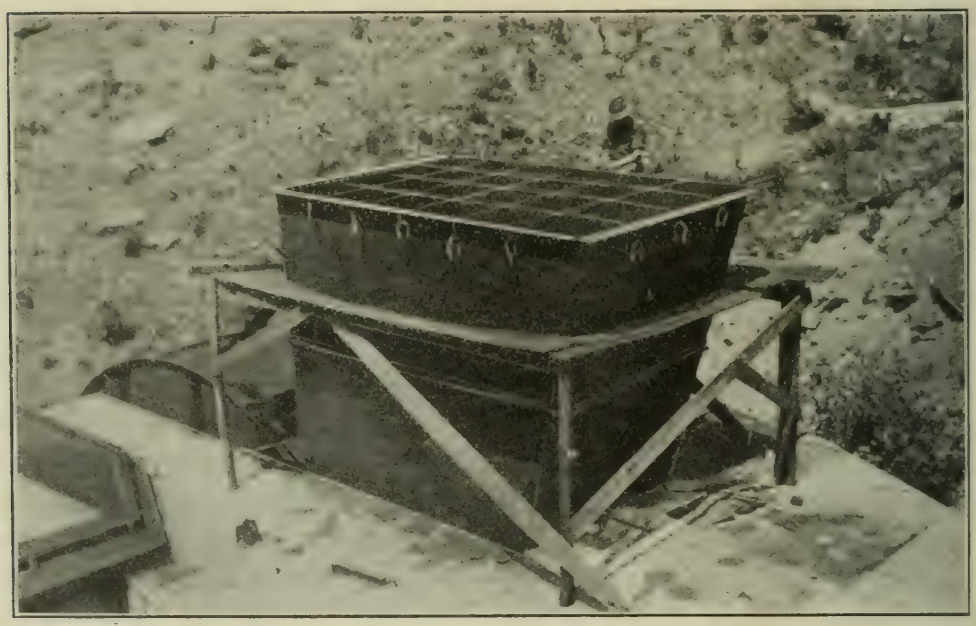


FIG. 12 DIVISION OF TOP OF CHIMNEY INTO EQUAL AREAS

27 In all of the tests on fans shown in the appendix, a differential manometer reading to 1/100 in. was used. This instrument used a special mineral oil as a working fluid but the angle of inclina-

tion was such as to cause it to read correct for water. Careful comparison with a direct-reading water manometer, proved that this angle was the proper one and that the instrument read correctly.

28 In the tests on headings a board was nailed to the wall of the cubby-hole and the manometer screwed to it, care being taken that this instrument was kept perfectly level. In taking measurements on the fan chimneys a board was fastened to the side of the chimney and the manometer fastened thereto as in the mine headings.

CONCLUSIONS

29 From our experience with the three types of Pitot tubes, we would draw the following conclusions:

a The Taylor tube, although probably very accurate and extremely sensitive on the air velocities for which it was designed, say up to 3000 ft. per min., is clumsy, difficult of manipulation, fragile, and altogether unsuited to high velocity measurement.

b Tubes of the Gebhardt type are light, compact, very portable, convenient and easy of manipulation. They are very accurate when used on velocities up to say, 6000 ft. per min. Beyond this point or beyond the limits of the common formula, their accuracy is somewhat problematical.

c The United States No. 1 tube is less convenient and portable than the Gebhardt. It is, however, much simpler in construction and will stand much more abuse. It does not require anywhere near as much care in manipulation as either of the other two types. It has been found accurate on air velocities as low as 600 ft. per min. The upper limit of its accuracy is unknown but it has been successfully used on velocities exceeding 25,000 ft. per min.

d The exact formula, as explained above, is much more accurate, especially on the higher velocities, than the one usually employed.

e Present data on the friction of air in pipes are largely in error. Our tests showed for the length and size of pipe used that the friction of the air was just about one-half what it was calculated to be. Additional accurate data along this line are sadly needed.

The writer wishes to acknowledge his indebtedness to Mr. Edward O'Toole, general superintendent of the United States Coal & Coke Company of Gary, W. Va., and to Mr. Howard N. Eavenson, chief engineer of the same company, under whose direct supervision the work above described was conducted. He also wishes to thank Mr. George C. Hicks of the P. H. & F. M. Roots Company, of Connersville, Ind., Capt. D. W. Taylor, naval constructor of the United States Navy Yard, Washington, D. C., and Mr. J. H. Klepinger, Great Falls, Mont., for valuable suggestions and aid.

APPENDIX

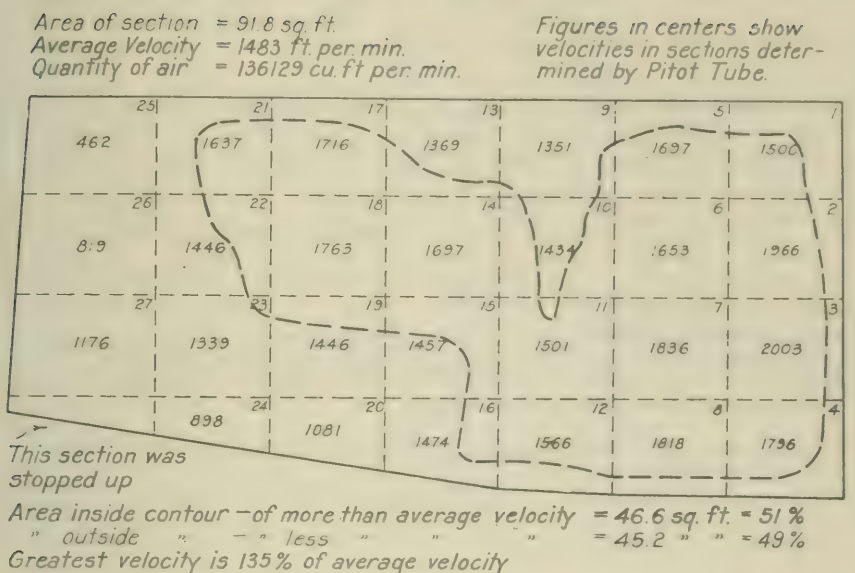

Area of section = 91.8 sq. ft.
Average Velocity = 1483 ft. per. min.
Quantity of air = 136129 cu. ft per. min.
Figures in centers show velocities in sections determined by Pitot Tube.
25 21 17 13 9 5 1
462 1637 1716 1369 1351 1697 1500
26 22 18 14 10 6 2
899 1446 1763 1697 1434 1653 1966
27 23 19 15 11 7 3
1176 1339 1446 1457 1501 1836 2003
24 20 16 12 8 4
898 1081 1474 1566 1818 1796
This section was stopped up
Area inside contour – of more than average velocity = 46.6 sq. ft. = 51 %
" outside " – " less " " " = 45.2 " " = 49 %
Greatest velocity is 135% of average velocity

FIG. 13 SECTION AND VELOCITIES IN AIRWAY, FAN TEST NO. 9

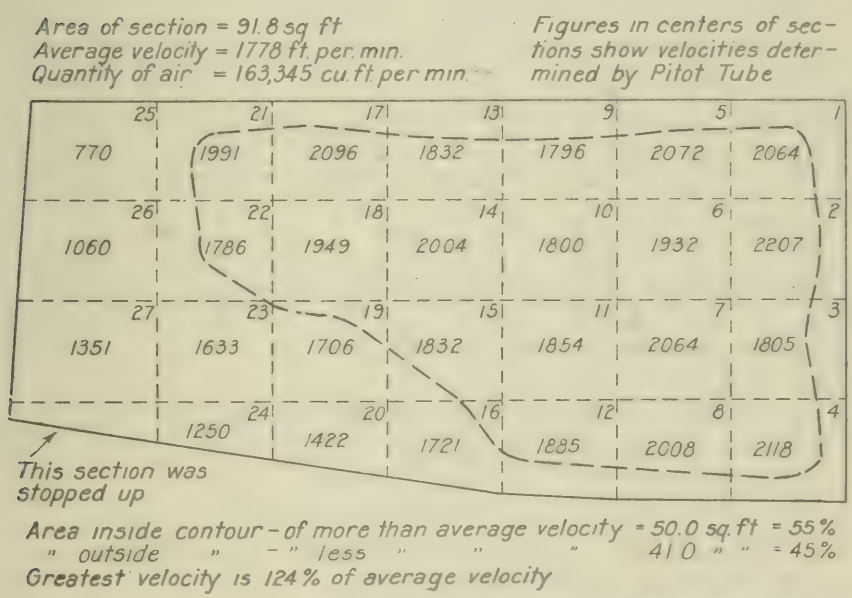

Area of section = 91.8 sq ft
Average velocity = 1778 ft. per. min.
Quantity of air = 163,345 cu. ft per min.
Figures in centers of sections show velocities determined by Pitot Tube
25 21 17 13 9 5 1
770 1991 2096 1832 1796 2072 2064
26 22 18 14 10 6 2
1060 1786 1949 2004 1800 1932 2207
27 23 19 15 11 7 3
1351 1633 1706 1832 1854 2064 1805
24 20 16 12 8 4
1250 1422 1721 1885 2008 2118
This section was stopped up
Area inside contour – of more than average velocity = 50.0 sq. ft = 55%
" outside " – " less " " " 41.0 " " = 45%
Greatest velocity is 124% of average velocity

FIG. 14 SECTION AND VELOCITIES IN AIRWAY, FAN TEST NO. 10

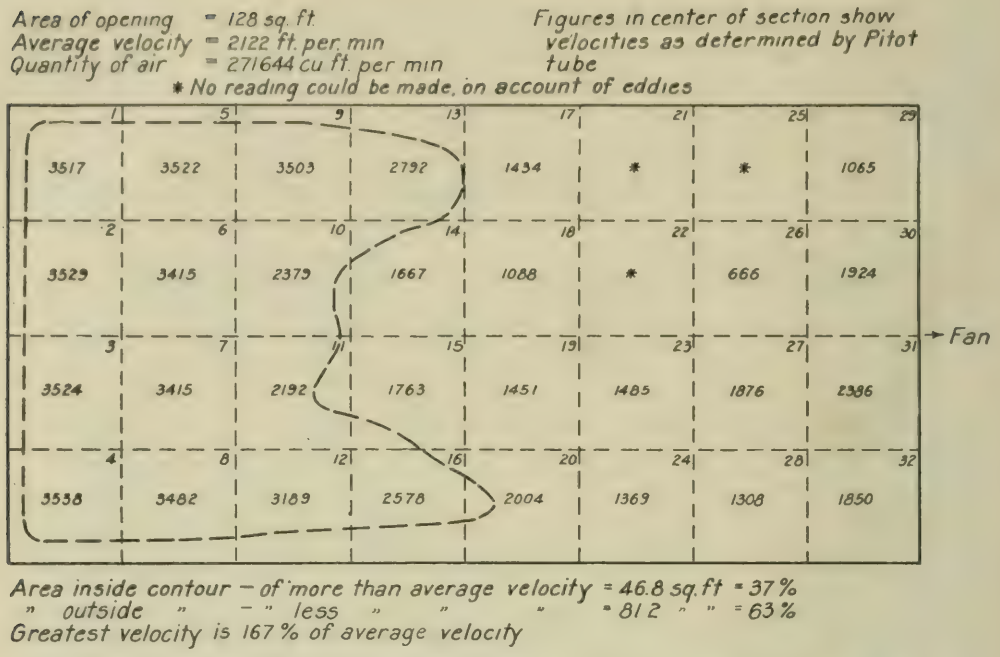

Area of opening = 128 sq. ft.
Average velocity = 2122 ft. per. min
Quantity of air = 271644 cu ft. per min
Figures in center of section show velocities as determined by Pitot tube
* No reading could be made, on account of eddies
1 5 9 13 17 21 25 29
3517 3522 3503 2792 1434 * * 1065
2 6 10 14 18 22 26 30
3529 3415 2379 1667 1088 * 666 1924
3 7 11 15 19 23 27 31
3524 3415 2192 1763 1451 1485 1876 2386
4 8 12 16 20 24 28 32
3538 3482 3189 2578 2004 1369 1308 1850
Fan
Area inside contour – of more than average velocity = 46.8 sq. ft = 37 %
" outside " – " less " " " = 81.2 " " = 63 %
Greatest velocity is 167 % of average velocity

FIG. 15 SECTION AND VELOCITIES AT TOP OF CHIMNEY, FAN TEST NO. 11

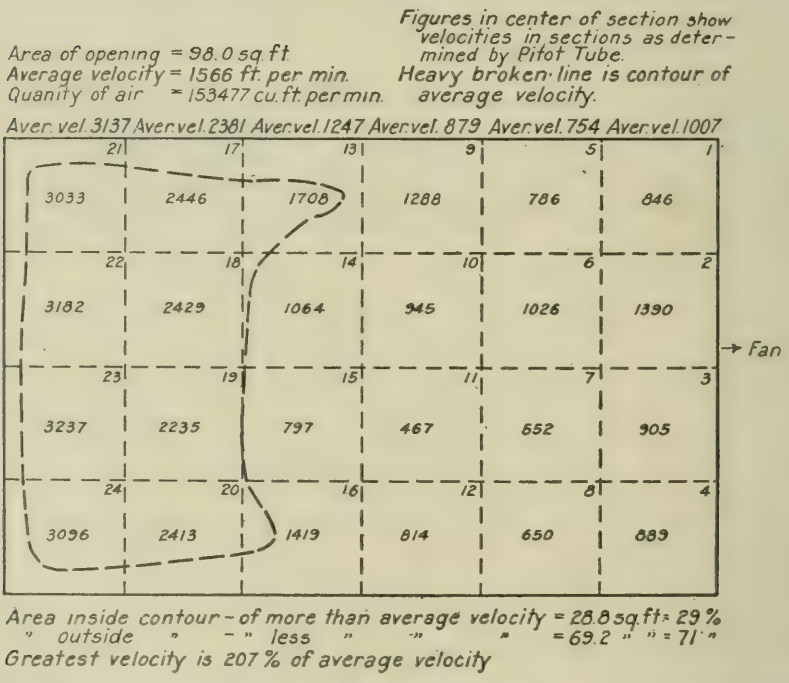

Area of opening = 98.0 sq. ft.
Average velocity = 1566 ft. per min.
Quanity of air = 153477 cu. ft. per min.
Figures in center of section show velocities in sections as determined by Pitot Tube.
Heavy broken line is contour of average velocity.
Aver. vel. 3137 Aver. vel. 2381 Aver. vel. 1247 Aver. vel. 879 Aver. vel. 754 Aver. vel. 1007
21 17 13 9 5 1
3033 2446 1708 1288 786 846
22 18 14 10 6 2
3182 2429 1064 945 1026 1390
23 19 15 11 7 3
3237 2235 797 467 652 905
24 20 16 12 8 4
3096 2413 1419 814 650 889
Fan
Area inside contour – of more than average velocity = 28.8 sq. ft = 29 %
" outside " – " less " " " = 69.2 " " = 71 "
Greatest velocity is 207 % of average velocity

FIG. 16 SECTION AND VELOCITIES AT TOP OF CHIMNEY, FAN TEST NO. 5

TABLE 2 TESTS OF BLOWER WITH 36-IN. PIPE BLANKED

No. 9 Works, United States Coal & Coke Company, April 10, 1910

	U. S. Pitot Tube No. 1								Gebhardt Pitot Tube No. 1.							
No. of Run	Vacuum in 36-in. Pipe	Dynamic Readings, 16-in. Pipe	Averages	Static Readings	Averages	Combined Averages	Weight of Air per Cu. Ft., Lb.	Velocity in 16-in. Pipe, Ft. per Minute	Dynamic Readings	Averages	Static Readings	Averages	Combined Averages	Weight of Air per Cu. Ft., 605	Velocity Air in 16-in. Pipe, Ft. per Minute	Blower, R.p.m.
	In. Mercury								In. Mercury							
1	2.40	1.20		3.00					1.10		3.30					
2	2.40	1.15		3.02					1.10		3.20					
3	2.40	1.25	1.24	3.03	3.03	1.79	0.0646	19410	1.20	1.17	3.20	3.22	2.05	0.0641	20710	....
4	2.40	1.40		3.07					1.25		3.20					
5	2.40	1.20		3.02					1.20		3.20					
6	2.40	1.30		3.00					1.10		3.10					
7	2.40	1.25		2.97					1.10		3.10					
8	2.40	1.15	1.22	3.00	2 99	1.77	0.0647	19290	1.10	1.11	3.16	3.14	2.03	0.0643	20540	87
9	2.40	1.20		2.96					1.10		3.12					
10	2.40	1.20		3.04					1 15		3 20					
11	2.40	1.30		3.02					1.15		3.05					
12	2.40	1.40		3.00					1.20		3 05					
13	2.40	1.35	1.35	3.00	3.00	1.65	0.0647	18740	1.25	1.19	3 03	3.05	1.86	0.0646	19730	86.5
14	2.40	1.30		2.97					1.20		3.07					
15	2.40	1.40		3.00					1.15		3.07					
16	2.40	1.45		2.97					1.30		3.00					
17	2.40	1.40		3.00					1.25		3.00					
18	2.40	1 40	1.42	3.00	2.98	1.56	0.0648	18280	1.25	1.27	2.98	3.00	1.73	0.0647	19110	87
19	2.40	1.40		2.97					1.30		3.03					
20	2.40	1.45		2.98					1.25		3.00					
21	2.40	1.55		2.94					1.55		2.70					
22	2.36	1.50		2.94					1.60		2.70					
23	2.36	1.45	1.47	2.94	2.93	1.46	0.0649	17740	1.60	1.59	2.70	2.70	1.11	0.0655	15600	87
24	2.36	1.45		2.91			Aver.	18692	1.60		2.70			Aver.	19138	
25	2.36	1.40		2.93					1.60		2.70					

Barometer, 28.64; temperature, 64 deg.; humidity, 41 per cent; mercury, 846.5.

$$V = 4041.6 \sqrt{\frac{p_2 - p_1}{P_1}} \ (1 - \text{etc.})$$

TABLE 3 TESTS OF BLOWER WITH 36-IN. PIPE BLANKED

No. 9 Works, United States Coal & Coke Company, April 10, 1910

	Gebhardt Pilot Tube No. 1								Taylor Pitot Tube No. 3							
No. of Run.	Vacuum in 36-In. Pipe	Dynamic Readings 16-In. Pipe	Averages.	Static Readings	Averages	Combined Averages	Velocity in 16-In. Pipe, Ft. per Minute	Weight of Air per Cu. Ft., Lb.	Dynamic Readings	Averages	Static Readings	Averages	Combined Averages	Weight of Air per Cu. Ft., Lb.	Velocity in 16-In. Pipe, Ft. per Minute	Blower, R. p. m.
	In. Mercury								In. Mercury							
1	2.40	1.10		3.75					1.00		2.17					
2	2.40	1.10		3.65					1.00		2.21					
3	2.40	1.10	1.10	3.65	3.66	2.56	23080	0.0631	0.95	0.99	2.20	2.18	1.19	0.0668	15670	87
4	2.40	1.10		3.65					0.95		2.18					
5	2.40	1.10		3.62					1.05		2.15					
6	2.40	1.25		3.55					1.13		2.17					
7	2.40	1.25		3.55					1.10		2.18					
8	2.38	1.20	1.22	3.55	3.55	2.33	22140	0.0633	1.15	1.16	2.18	2.19	1.03	0.0668	14780	87
9	2.40	1.20		3.55					1.25		2.21					
10	2.40	1.20		3.55					1.15		2.19					
11	2.36	1.15		3.55					1.45		2.29					
12	2.40	1.15		3.55					1.35		2.27					
13	2.36	1.20	1.19	3.52	3.53	2.34	22150	0.0634	1.25	1.32	2.20	2.25	0.93	0.0666	14150	87
14	2.36	1.20		3.50					1.30		2.20					
15	2.36	1.25		3.55					1.25		2.29					
16	2.36	1.40		3.40					1.35		2.27					
17	2.36	1.40		3.35					1.40		2.24					
18	2.36	1.40	1.40	3.35	3.36	1.96	20460	0.0638	1.35	1.35	2.24	2.24	0.89	0.0666	13880	87.5
19	2.36	1.40		3.35					1.30		2.24					
20	2.36	1.40		3.35					1.35		2.21					
21	2.40	1.65		2.93					1.35		2.21					
22	2.38	1.65		2.95					1.45		2.24					
23	2.40	1.65	1.65	2.92	2.93	1.28	16760	0.0649	1.50	1.44	2.27	2.26	0.82	0.0666	13390	87
24	2.36	1.65		2.92					1.45		2.29					
25	2.40	1.65		2.92		Aver.	20918		1.45		2.27			Aver.	14371	87.1

Barometer 28.64; temperature, 64 deg.; humidity, 41 per cent; mercury, 846.5.

$$V = 4041.6\sqrt{\frac{p_2 - p_1}{P_1}}\,(1 - \text{etc.})$$

TABLE 4 TESTS OF BLOWER FOR LEAKAGE, 16-IN. PIPE OPEN, OTHERS CLOSED

No. 9 Works, United States Coal & Coke Company, April 10, 1910

	U. S. Pitot Tube No. 1								Gebhardt Pitot Tube No. 1							
No. of Run.	Vacuum in 36-in. Pipe	Dynamic Readings 16-in. Pipe	Averages	Static Readings	Averages	Combined Averages	Weight of Air per Cu. Ft., Lb.	Velocity in 16-in. Pipe, Ft. per Min.	Dynamic Readings	Averages	Static Readings	Averages	Combined Averages	Weight of Air per Cu. Ft., Lb.	Velocity of Air in 16-in. Pipe, Ft. per Min.	Blower, R. p. m.
	In. Mercury								In. Mercury							
1	1.60	0.58		1.70					0.65		1.80					
2	1.60	0.63		1.75					0.60		1.80					
3	1.56	0.58	0.60	1.73	1.72	1.12	0.0680	14820	0.60	0.61	1.85	1.85	1.24	0.0676	15590	.....
4	1.56	0.58		1.71					0.60		1.90					
5	1.56	0.63		1.71					0.61		1.90					
6	1.56	0.75		1.78					0.65		1.70					
7	1.60	0.80		1.77					0.65		1.85					
8	1.60	0.75	0.77	1.75	1.77	1.00	0.0679	14210	0.65	0.65	1.80	1.80	1.15	0.0677	15080	87.5
9	1.60	0.80		1.78					0.65		1.83					
10	1.56	0.75		1.78					0.65		1.80					
11	1.56	0.80		1.78					0.70		1.80					
12	1.56	0.85		1.78					0.70		1.80					
13	1.56	0.85	0.82	1.78	1.78	0.96	0.0679	13980	0.75	0.71	1.78	1.79	1.08	0.0678	14690	87.5
14	1.58	0.80		1.78					0.72		1.78					
15	1.56	0.80		1.78					0.70		1.78					
16	1.56	0.90		1.78					0.72		1.75					
17	1.60	0.90		1.78					0.72		1.70					
18	1.60	0.90	0.90	1.78	1.78	0.89	0.0679	13470	0.72	0 72	1.75	1.74	1.02	0.0679	14300	87.5
19	1.60	0.95		1.78					0.72		1.75					
20	1.60	0.85		1.78					0.72		1.75					
21	1.60	0.95		1.78					0.95		1.48					
22	1.56	1.00		1.78					0.95		1 48					
23	1.56	0.95	0.95	1.78	1.78	0.83	0.0679	13140	0.95	0.95	1.48	1.48	0.53	0.0686	10610	86.5
24	1.56	0.95		1.78			Aver.	13924	0.95		1.48			Aver.	14054	87.2
25	1.56	0.90		1.78					0.95		1.48					

Barometer 28.64; temperature, 64 deg.; humidity, 41 per cent; mercury, 846.5

$$V = 4041.6 \sqrt{\frac{p_2 - p_1}{P_1}} (1 - \text{etc.})$$

TABLE 5 DATA FOR FAN TEST NO. 5

No. 4 Works, United States Coal & Coke Company, January 28, 1911

Section Number	Area, Sq. Ft.	Pitot Tube			Anemometer	
		Reading, In. Water	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.
1	4.08	0.0447	846	3452	1150	4692
2	4.08	0.1207	1390	5671	1680	6854
3	4.08	0.0511	905	3692	890	3631
4	4 08	0.0493	889	3627	1190	4855
5	4.08	0.0386	786	3207	895	3652
6	4.08	0.0657	1026	4186	1120	4570
7	4.08	0 0190	552	2250	690	2815
8	4.08	0.0264	650	2652	770	3142
9	4.08	0.1036	1288	5255	1400	5712
10	4.08	0.0558	945	3856	1170	4774
11	4.08	0.0136	467	1905	840	3427
12	4.08	0.0414	814	3321	1080	4406
13	4.08	0.1821	1708	6969	1810	7385
14	4.08	0.0707	1064	4341	1290	5263
15	4 08	0 0397	797	3252	1065	4345
16	4.08	0.1257	1419	5790	1780	7262
17	4.08	0.3737	2446	9980	3100	12648
18	4.08	0.3686	2429	9910	3080	12566
19	4.08	0.3118	2235	9119	2780	11342
20	4.08	0.3636	2413	9845	3090	12607
21	4 08	0.5743	3033	12375	3760	15341
22	4.08	0.6321	3182	12983	4095	16708
23	4 08	0.6543	3237	13207	4050	16524
24	4.08	0.5984	3096	12632	3765	15361
	98.00		1566	153477	1937	189882

Conditions: Fan on mine, but doors at back of fan open, allowing all air possible to enter fan.

Air: Measured at top of chimney. Beginning Ending Barometer 28.94

Temperature outside, 44 deg. from chimney..... 46½ deg. 53½ deg.

Humidity outside, 71 per cent from chimney.... 72 per cent 58 per cent $V = 4002\sqrt{h}$

Weight of air per cu. ft., 0.0762 lb..... 0.0758 0.0747

Average weight of air from chimney, 0.0752 lb. per cu. ft.

TABLE 6 DATA FOR FAN TEST NO. 9

No. 6 Works, United States Coal & Coke Company, February 18, 1911

Section Number	Area, Sq. Ft.	Pitot Tube			Anemometer	
		Reading, In. Water	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.
1	3.49	0.138	1506	5256	2213	7723
2	3.52	0.235	1966	6920	2475	8712
3	3.55	0.244	2003	7111	2490	8839
4	3.53	0.196	1796	6340	2485	8772
5	3.44	0.175	1697	5838	2508	8628
6	3.50	0.166	1653	5785	2445	8557
7	3.50	0.205	1836	6426	2402	8407
8	3.36	0.201	1818	6108	2368	7956
9	3.40	0.111	1351	4593	2345	7973
10	3.50	0.125	1434	5019	2225	7787
11	3.50	0.137	1501	5253	2260	7910
12	3.24	0.149	1566	5074	2355	7630
13	3.38	0.114	1369	4627	2300	7774
14	3.50	0.175	1697	5939	2438	8533
15	3.50	0.129	1457	5100	2313	8095
16	2.86	0.132	1474	4216	2207	6312
17	3.38	0.179	1716	5800	2615	8839
18	3.50	0.189	1763	6170	2464	8624
19	3.50	0.127	1446	5061	2286	8001
20	2.24	0.071	1081	2421	2065	4626
21	3.40	0.163	1637	5566	2375	8075
22	3.50	0.127	1446	5061	2225	7788
23	3.50	0.109	1339	4687	2175	7612
24	1.62	0.049	898	1455	1875	3037
25	3.86	0.013	462	1783	1400	5404
26	4.20	.....	819*	3440	1055	4431
27	4.32	0.084	1176	5080	1925	8316
28	....	....	....	....	....	....
	91.79		1483	136129	2226	204361

* Average velocity of Nos. 25 and 27. Could not obtain reading on account of eddies.

Conditions: normal, fan on mine. Temperature, outside 57 deg. fahr. Temperature, inside 53½ deg. fahr.; humidity, outside 88 per cent, inside 100 per cent; barometer, 28.46, weight of inside air per cu. ft. 0.0732 lb., for Pitot tube $V = 4056\sqrt{h}$.

Measurement made in airway leading to fan section was divided by strings into approximately equal parts. Section No. 28 was stopped up, as it was very small and at the bottom.

Readings taken by Pitot tube and anemometer held in center of each section by a stand. Readings taken for period of one minute in each position. Could get no reading by tube in section No. 26, evidently on account of eddy currents; velocity given is average of those in the two adjoining sections. Fan running exhausting. Airway makes a slight turn a short distance beyond the section, causing velocities to be higher on one side than on the other.

TABLE 7 DATA FOR FAN TEST NO. 10

No. 6 Works, United States Coal & Coke Company, February 18, 1911

Section Number	Area, Sq. Ft.	Pitot Tube			Anemometer	
		Reading, In. Water	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.
1	3.49	0.259	2064	7203	2940	10261
2	3.52	0.296	2207	7769	3082	10849
3	3.55	0.198	1805	6409	2684	9528
4	3.53	0.273	2119	7480	3160	11155
5	3.44	0.261	2072	7128	2680	9219
6	3.50	0.227	1932	6762	2580	9030
7	3.50	0.259	2064	7224	2635	9222
8	3.36	0.245	2008	6747	2745	9223
9	3.40	0.196	1796	6106	2330	7922
10	3.50	0.197	1800	6300	2465	8627
11	3.50	0.209	1854	6489	2450	8565
12	3.24	0.216	1885	6107	2490	8068
13	3.38	0.204	1832	6192	2705	9143
14	3.50	0.244	2004	7014	2765	9677
15	3.50	0.204	1832	6412	2490	8715
16	2.86	0.180	1721	4922	2355	6735
17	3.38	0.267	2096	7084	2660	8991
18	3.50	0.231	1949	6821	2565	8977
19	3.50	0.177	1706	5971	2430	8505
20	2.24	0.123	1422	3185	2343	5248
21	3.40	0.241	1991	6769	2629	8939
22	3.50	0.194	1786	6251	2455	8592
23	3.50	0.162	1633	5715	2400	8400
24	1.62	0.095	1250	2025	2110	3418
25	3.86	0.036	770	2972	2000	7720
26	4 20	.....	1060	4452	1675	7035
27	4.32	0.111	1351	5836	2245	9698
	91.79		1778	163345	2521	231462

Conditions: fan on mine-doors on main heading between 1st and 2d return and on 1st return open, allowing air to short circuit through mine. Temperature inside 53½ deg. fahr.; humidity. inside 100 per cent; barometer 28.46; for Pitot tube $V = 4056\sqrt{h}$.

Same conditions of measurement as test No. 9.

TABLE 8 DATA FOR FAN TEST NO. 11

No. 6 Works, United States Coal & Coke Company, February 18, 1911

Section Number	Area, Sq. Ft.	Pitot Tube			Anemometer	
		Reading, In. Water	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.	Velocity, Ft. per Min.	Quantity, Cu. Ft. per Min.
1	4.0	0.752	3517	14068	5414	21656
2	4.0	0.757	3529	14116	3920	15680
3	4.0	0.755	3524	14096	3696	14784
4	4.0	0.761	3538	14152	4140	16560
5	4.0	0.754	3522	14088	3564	14256
6	4.0	0.709	3415	13660	3058	12232
7	4.0	0.709	3415	13660	3132	12528
8	4.0	0.737	3482	13928	3374	13496
9	4.0	0.746	3503	14012	3392	13568
10	4.0	0.344	2379	9516	2468	9872
11	4.0	0.292	2192	8768	2536	10144
12	4.0	0.618	3189	12756	3324	13296
13	4.0	0.474	2792	11168	2890	11560
14	4.0	0.169	1667	6668	2090	8360
15	4.0	0.189	1763	7052	2060	8240
16	4.0	0.404	2578	10312	2816	11264
17	4.0	0.125	1434	5736	1560	6240
18	4.0	0.072	1088	4352	1540	6160
19	4.0	0.128	1451	5804	1920	7680
20	4.0	0.244	2004	8016	2060	8240
21	4.0	.....	.....	.....	476	1904
22	4.0	.....	.....	.....	760	3040
23	4.0	0.134	1485	5940	1770	7080
24	4.0	0.114	1369	5476	1630	6520
25	4.0	.....	....	.....	310	1240
26	4.0	0.027	666	2664	830	3320
27	4.0	0.214	1876	7504	1960	7840
28	4.0	0.104	1308	5232	1570	6280
29	4.0	0 069	1065	4260	1250	5000
30	4.0	0.225	1924	7696	2070	8280
31	4.0	0.346	2386	9544	2450	9800
32	4.0	0.208	1850	7400	2000	8000
	128.0		2122	271644		304120

Conditions: doors to fan and doors inside mine all open, allowing all air possible to reach fan. Air measured at top of chimney. Temperature, 53½ deg. fahr.; humidity, 100 per cent; barometer, 28.46; for Pitot tube, $V = 4056 \sqrt{h}$.

Measurement made on top of chimney of fan. Area was divided by strings into 32 equal parts and readings were taken in center of each section for periods of 30 seconds. Both anemometer and tube were held in plane of top of chimney. Pitot tube readings could not be obtained in sections Nos. 21, 22 and 25, on account of eddies. These were very noticeable, as the anemometer in the same sections would stop, reverse itself, stop, run properly, etc.

DISCUSSION

G. F. GEBHARDT[1] (written). In Par. 29*b* Mr. Kneeland states that tubes of the Gebhardt type are very accurate up to velocities of 6000 ft. per min., and beyond this their accuracy is somewhat problematical.

Since the Gebhardt tubes purchased by Mr. Kneeland were not calibrated for velocities above 8000 ft. per min., the writer is obliged to base the performance of these instruments for higher velocities upon Mr. Kneeland's own figures.

According to the data in Tables 2 and 4, the average velocity for the Gebhardt tube is about 8/10 of 1 per cent higher than that for the United States tube at a velocity of approximately 14,000 ft. per min., and about 2.8 per cent higher at a velocity of 19,500 ft. per min. This variation is extremely small when we consider that the average velocities are based on readings taken at only five points in a 16-in. pipe.

It is to be regretted that Mr. Kneeland did not make complete traverses of the pipe and take readings at more points along the line of traverse. Calculations based upon a few observations on one side of a pipe are apt to lead to erroneous conclusions.

The velocity curves for the United States tubes in Figs. 7, 8 and 9, differ greatly from the usual characteristics for the flow of air in pipes. Curves plotted from a large number of scattering tests are in accord with the characteristic of the Gebhardt tube as shown in these figures.

The method of calculating the average velocity as described in Par. 11, necessitates dividing the cross-section of the pipe into a number of annular rings of equal areas. The opening of the dynamic tube is placed at the middle of the annular ring and the velocity indicated by the tube is assumed to be uniform over the entire area of the ring.

A more accurate method and one which makes it unnecessary to place the dynamic nozzle at a predetermined point in the pipe is described by R. Burnham,[2] and by Prof. Carl C. Thomas.[3]

[1] Prof. Mechanical Engrg., Armour Inst. of Tech., Chicago, Ill.

[2] Engineering News, December 21, 1905, p. 660.

[3] Journal, Franklin Institute, November 1911, p. 411.

The curves in Fig. 17 are plotted according to the Burnham method from the data given in Table 2 of the present paper. It will be noted that the mean velocity for the Gebhardt tube is 18,750 ft. per min., and for the United States tube 18,550 ft. per min., a difference of about 1 per cent, as against 2.8 per cent by the annular ring method. Further experiment is necessary to prove conclusively whether this discrepancy is due to the difference in indications of the instruments or to the error in basing the mean velocity upon a few readings at widely separated points.

The writer wishes to state that the so-called Gebhardt tube originated with R. Burnham, former professor of experimental engineering at Armour Institute of Technology.

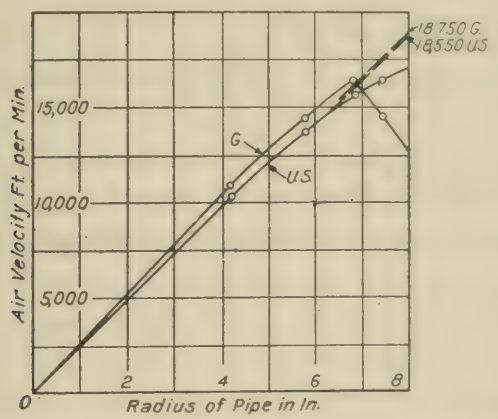


FIG. 17 RADIUS OF PIPE AND AIR VELOCITY DIAGRAM

R. C. CARPENTER said that during the last eight or ten years the Pitot tube had probably been studied more in connection with engineering work than any other measuring instrument. As far back as 1884 he remembered a German article which pointed out the fact that the results obtained by the Pitot tube for measuring air were much more reliable than those obtained generally by an anemometer. The popularity and accuracy of the tube is increasing.

Some difficulty has been found in using tubes of different kinds for measuring static pressure. Tubes of almost all sorts, so far as the impact and nozzle end are concerned have been tried; practically the same results as to impact head have been obtained with various forms of tubes. A plain tube with an opening on one side is quite accurate for giving the impact head and for most purposes, and on account of its convenience is much to be preferred over other

types. A tube 2/16 in. in diameter is as good as a tube ½ in. in diameter, as to results.

There is more difficulty, however, in measuring static pressure. For this purpose the Taylor tube is provided with a round chamber in which slots are cut, which surrounds the impact tube. It is an awkward tube to use, although it is considered accurate for static pressure, and yet he was quite positive that in a great many cases it fails to give the correct static pressure. The problem of getting impact pressures with the Pitot tube may be considered as practically solved, and essentially the same results may be obtained with almost any form of tube. This is far from being the case in getting static pressure.

For securing static pressures he favored the use of a large tube placed in the wall itself, and protected with baffle plates from any cross-current, since he felt that this was more reliable than any tube placed where it was likely to be hit by an eddy-current in the pipe.

W. H. CARRIER thought that a few more data should be given with reference to the amount of air handled by the rotary blower, and as to how the amount of air was determined. He did not understand whether it was the capacity measured at atmospheric pressure that was given or volume under the vacuum shown in the test. Apparently it is the volume under vacuum that is given, but this is not specifically stated. In order to accept these results sufficient data should be given so that the reader might draw his own conclusions.

D. W. TAYLOR[1] (written). The experimental results given by Mr. Kneeland cannot be regarded as acceptable to those who, like myself, are firm believers in the accuracy of Pitot tube methods, provided proper Pitot tubes are used. The results are so discordant that we must conclude either that Pitot tube methods are unreliable or that under the circumstances of use some of the Pitot tubes were inaccurate. I believe the latter conclusion is the correct one.

There is only one real problem to be solved in designing an accurate pitot tube. As regards the impact end, or dynamic nozzle as the author calls it, there is no difficulty at all. It is now well known that almost any tube placed with its axis parallel to the flow of a fluid, so that the fluid is brought to rest in front of the opening will accu-

[1] Naval Constructor, U.S.N., Washington, D. C.

rately transmit the dynamic head. When we come to the static side, however, the problem is entirely different. We wish to determine the pressure in the moving fluid without changing the pressure or velocity of the latter. At the same time the fluid in the pipe leading from the static nozzle is necessarily at rest, so the problem is to transmit accurately the pressure in the moving fluid to the fluid at rest without changing the velocity or pressure of the latter.

To accomplish this we must avoid openings so arranged that there is an impact action which would make the static pressure greater than the true pressure in the moving fluid; and, on the other hand, we must avoid designs by which there is an aspiration effect around the static nozzle, otherwise the velocity and pressure of the moving fluid at the static nozzle are materially changed by the disturbance caused by the pitot tube itself.

It is true that in the Gebhardt tube the static opening is purposely and very ingeniously arranged so that there is a certain amount of impact pressure intended to neutralize the aspiration effect. It seems very probable, however, that the angle of bevel of the static nozzle by which this result is accomplished would vary with the velocity and with the fluid. My understanding is that this tube was put out by Professor Gebhardt for use in steam after careful scientific investigations of its accuracy for this use. It is evident that a fluid such as air under a pressure of 14 or 15 lb. to the sq. in. would act differently with respect to the static opening from a fluid such as steam under a pressure of 100 lb. or so to the sq. in. The transmission of pressure into the static opening must be effected by the specific gravity and head of the moving fluid.

It is not at all clear to me from Fig. 5 just how the static nozzle *B* of the United States tube is constructed, or more particularly, just how the air would pass across the opening, but if this nozzle were secured into the wall of a pipe without projecting inside, it does not seem to me that there would be necessity for any particular type of nozzle. The hole through the wall of the pipe would act as the nozzle.

The author appears to have had difficulty in handling the Pitot tube such as we have used in testing ventilating fans, but as he points out, this tube was never intended for high velocities. Still, it seems to me that the use for high velocities is entirely a question of the supports which have little or no influence upon the accuracy of the tube.

Apparently the author attempted to apply the principles of this

design to more substantial tubes which were made larger. For high velocity work my tendency would be to make the tube distinctly smaller.

With reference to the accuracy of the several types of tubes I have already pointed out that this is in substance entirely a question of the accuracy of the static nozzle. In Tables 2, 3 and 4 the agreement on the impact side of the several types of tubes is fairly good; the discrepancies come on the static side.

In this connection I may quote some remarks I made in 1905 before the Society of Naval Architects and Marine Engineers upon the subject of the type of tube used for experiments with the ventilating fans and pipes.[1]

> I very early satisfied myself by a simple experiment that this type of tube would record the side pressure accurately. A strong blast of air was forced through a 9-in. pipe from a fan, and the Pitot tube arranged so it could be moved back and forth lengthwise in the center of the jet from the pipe, readings being taken of the pressure recorded through the side opening. When the tube was very close to the mouth of the pipe an appreciable pressure above that of the atmosphere was shown, as was to be expected. As the tube was moved back from the mouth of the pipe, although still in a very strong and well-defined jet of air, the pressure fell off until it exceeded the atmospheric pressure by a very small amount only. Now, it was evident that in the center of such a jet the pressure at the mouth of the pipe must be appreciably above that of the atmosphere, but at a comparatively short distance beyond the mouth of the pipe the pressure in the center of the jet must fall very nearly to that of the undisturbed atmosph re although it probably never quite reaches it. If the Pitot tube had been in error, as used to be supposed, the pressure recorded in the center of the free jet would have been materially below that of the atmosphere, while as a matter of fact it never quite reached the atmospheric pressure, agreeing entirely with theory.

Since this was written I have learned of a number of instances where the type of tube referred to gave results that were practically exact. A very recent one was in a paper[2] by Prof. Carl C. Thomas, describing certain experiments made at the University of Wisconsin.

The quantity of air passing through a 16-in. pipe was measured simultaneously by three different methods, namely: a pitot tube substantially identical with the Taylor type as described by Mr. Kneeland; an electric meter apparatus recently brought out by Professor Thomas which is exceedingly accurate; and a venturi meter whose accuracy is well known. The three methods gave substan-

[1] Trans. Soc. Naval Arch. & Marine Engrs., vol. 13, p. 11.

[2] Journal, Franklin Institute, November 1911.

tially identical results, the velocities being, however, not over 5000 or 6000 ft. per min.

In Fig. 8, Mr. Kneeland gives velocities across a 16-in. pipe determined by two types of tubes and also velocity by blower. It appears from the text that for determining the velocity by blower he used the curve given in Fig. 3.

I think some additional information is needed in this connection. The blower when working under no pressure is stated to displace 300 cu. ft. per revolution. Fig. 3 shows that when working with an 8 in. vacuum it would displace 260 cu. ft. per revolution. Apparently, the 260 cu. ft. refers to free air, not the displacement of actual rarefied air.

If we take the ordinary barometer at 30 in. and assume that there is 8 in. of vacuum on each side of the Root blower it would presumably displace 300 cu. ft. of this rarefied air per revolution, equivalent to 220 cu. ft. of free air at 30-in. pressure. If this is the case it is difficult to understand how this blower drawing against an 8-in. vacuum and discharging against the pressure of the air could possibly displace a quantity of air per revolution equivalent to 260 cu. ft. of free air. Presumably the displacement would be something materially less than 220 cu. ft. Possibly Fig. 3 is intended to refer to rarefied air and I am in error in understanding the text otherwise.

It will be observed in Fig. 8 that a comparatively small reduction in the velocity as estimated from the blower would cause a close agreement between the blower velocity and that obtained by the Taylor No. 1 tube.

I am inclined to think that there was some aspiration effect in the other types of tubes used tending to lower the static pressures required and causing the apparent velocities to be too great. The results given in Fig. 11 rather confirm me in this view. The loss of pressure due to friction is just about 50 per cent of that given in Sturtevant's tables, which, of course, apply to sheet-iron pipes such as are used in ventilation.

In experiments made some years ago and described fully in a paper[1] before the Society of Naval Architects and Marine Engineers I carefully investigated the frictional resistance of sheet-iron pipes ranging in diameter from 6 in. to 27 in. and found that in the best cases the loss due to friction might be only some 80 per cent of that given by Sturtevant.

[1] Trans. Soc. Naval Arch. & Marine Engrs., vol. 13, p. 9.

I am very skeptical, however, of the possibility of pipes smooth enough to reduce the friction to but 50 per cent of that given by Sturtevant and am inclined to think that such results were obtained by overestimating the velocity in the pipes.

We have had a great deal of experience in the Navy with ventilation systems since 1905 and certainly have no reason to think that the coefficients of friction given then were too large.

THE AUTHOR. In reply to Mr. Gebhardt's criticism, we began our tests by taking several points of observation along one axis and an equal number along the axis at right angles to the first. The number of observation points was then gradually decreased to five along one radius, without changing the result noticeably. The last number was then adopted as our standard.

Both Mr. Carrier and Captain Taylor question the accuracy of Fig. 3. The name of this curve was, perhaps, unfortunate, as it shows rather the leakage of the blower than its capacity.

In reply to Captain Taylor's criticism it would appear to the writer that the accuracy of the static nozzle of the Taylor tube depends entirely upon the air passing along the side of the tube and across the static opening, exactly parallel to the axis of the tube. It would seem reasonable to suppose that the disturbance created in the air by any Pitot tube would be exactly analogous and similar to the disturbance created in the water by a moving boat, or other object. It is well-known that the amount of these disturbances is dependent both upon the shape or "lines" of a boat, and its velocity. A vessel which at a speed of, say 5 knots, will create little or no disturbance, may create a very large disturbance at, say 15 to 20 knots. It would seem altogether probable, therefore, that to secure accurate results with Pitot tubes of the Taylor type at high velocity, the "bow" or nose of the tube must be so designed that, at the velocity to be measured, the air will pass down the length of the tube and across the static opening, exactly parallel to the axis, remaining meanwhile at constant, or very nearly constant density throughout its passage. That this was the case with tubes used in our experiments, which were the identical tubes loaned us by Captain Taylor, the writer is very much inclined to doubt.

Captain Taylor makes reference to the results obtained in certain experiments at the University of Wisconsin by Prof. Carl C. Thomas. An inspection of the drawing of the Pitot tube used by Professor Thomas shows clearly that its dynamic nozzle was decidedly longer

and more slender than the same part of the tube used by us. In other words, the tube used by Professor Thomas had decidedly finer bow lines, and, therefore, created much less of a disturbance in the air along the cylindrical sides of the tube than did those used in the experiments above described. Furthermore, as Captain Taylor admits, the velocities employed were only 5000 to 6000 ft. per minute, while we were dealing with velocities as high as 20,000 to 25,000 ft. per minute, and, in some cases, even higher.

In considering the results obtained on the friction of air in the pipe line it should be borne in mind that these were the actual figures obtained on tests of a 24-in. pipe line over 1100 ft. long. That these results should be at variance with friction losses calculated from observations taken on smaller diameters or short lengths of pipe, is not in the least surprising.

No. 1343

NECROLOGY

MORRIS LANDA ABRAHAMS

Morris Landa Abrahams died on March 28, 1911, at San Antonio, Texas. Mr. Abrahams was born on March 21, 1885, at Austin, Texas, and after a preparatory education at the Hogsett Military Academy, Danville, Ky., and the Agricultural and Mechanical College of Texas, entered Cornell University, from which he was graduated in 1905 with the degree of M. E. Previous to his matriculation at Cornell he had acquired practical experience with the Deming Company, Salem, Ohio, in their drafting room, and with the Dean Brothers Steam Pump Works, Indianapolis, Ind., as assistant to the foreman of the testing floor. In 1907 he entered the employ of Walter S. Timmis, consulting engineer, New York City, where he engaged in power plant design, printing plant layouts, the development of a patent elevator safety device, heating and ventilation, and other similar phases of mechanical engineering. Later in the same year he left Mr. Timmis to become a member of the engineering force of the Victor Talking Machine Company in Camden, N. J. In 1910 he entered the service of the Government as assistant chief inspector attached to the Canal Commission, with headquarters at Washington, and retained this position until shortly before his death, when illness obliged him to resign his duties.

LOUIS R. ALBERGER

Louis R. Alberger was born April 10, 1864, at Buffalo, N. Y., and was educated at the public schools of that city. He studied engineering chemistry at the Scheffield Scientific School, Yale University, and later joined his father in the firm of Alberger and Salt, builders of salt-making apparatus in which the vacuum system was incorporated. This brought him into close contact with the subject which he made his life study, and after superintending for thirteen years the condensing department of Henry R. Worthington, he incorporated the Alberger Condenser Company, in 1901, and later the Alberger Pump Company, and acquired the controlling interest in the New-

burgh Ice Machine and Engine Company, Newburgh, N. Y. At the time of his death he was president of the Alberger Condenser Company.

Mr. Alberger was a member of the American Institute of Mining Engineers, the National Electric Light Association, the American Association for the Advancement of Science, Verein deutscher Ingenieure, and of a number of social and athletic organizations. He died in New York, January 31, 1911.

PAUL RAYMOND BROOKS

Paul Raymond Brooks was born in Chicago, Ill., August 17, 1877, and was educated at the Chicago Manual Training School and the Massachusetts Institute of Technology, from which he was graduated with the degree of B.S. in 1900. Following his graduation he entered the employ of the Chicago, Burlington & Quincy Railroad, where he began in the locomotive shop as an apprentice and worked his way through successive promotions to the position of acting foreman of the Burlington roundhouse of the company. In 1904 he became Western mechanical editor for the McGraw Publishing Company, New York. The following year he became a sales engineer for the Railway Appliances Company and the Otto Gas Engine Works, and in 1908 general sales manager for the Machine Sales Company, Peabody, Mass. In 1909 he associated himself with the Union Bag & Paper Company, Sandy Hill, N. Y., as mechanical engineer, and at the time of his death on March 11, 1911, was president and superintendent of the Del Monte Irrigation Company, Texas.

During his whole life Mr. Brooks took an active interest in the work of the Naval Reserves, and some time before his death had obtained first-class Government papers entitling him to serve as chief engineer on vessels of any tonnage during time of war.

ALEXANDER E. BROWN

Alexander E. Brown was born in Cleveland, December 14, 1852, and was educated at grammar and high schools there. He was graduated from the Brooklyn Polytechnic Institute in 1872 as a civil engineer. Immediately after graduation Mr. Brown joined the U. S. Geological Survey under F. P. Haven. Until November 1872 he was engaged in the exploration and location of the Yellowstone National Park. During the next two years he served as chief engineer of the Massillon Iron Bridge Company, Massillon, O., where he obtained his

first real experience in the iron business. From 1875 to 1879 he was engaged in bridge and building construction as engineer, designer and contractor. He later acted as superintendent at iron mines in the Lake Superior iron region and gained experience also in blast furnace work. In 1875 he invented and patented new machines and processes for annealing malleable iron castings, which are still in use at the National Malleable Iron Castings Company. In 1878 Mr. Brown became mechanical engineer for the Brush Electric Company then known as the Cleveland Electric Supply Company. While with this concern he developed the process and method of manufacturing light carbons. Still later he worked with Chas. F. Brush in developing and placing on the market the Brush arc lighting system. In 1879, Mr. Brown took up the problem of the rapid and economical unloading and handling of iron ore from boats. A year later he took out patents on the Brown hoisting and conveying machines erected on the docks of the N. Y., P. & O. Ry. Co., now part of the Erie system. He then organized the Brown Hoisting and Conveying Machinery Company for the manufacture of his machines. For the past 30 years Mr. Brown spent most of his time designing and manufacturing machinery for the handling of different machines. He had taken out several hundred patents on his various inventions.

He took an active interest in the development of trade schools in Cleveland, and was one of the directors of the Training School Company, organized in 1885, to open a carpenter shop for boys to which pupils were to be admitted free for a course extending over three years. He was a member of the American Institute of Mining Engineers, the Society of Naval Architects and Marine Engineers, Civil Engineers Club, Electric Engineers Club of Cleveland, the Engineers Club of New York and the Chamber of Commerce of Cleveland. He died April 26, 1911.

HENRY W. BULKLEY

Henry W. Bulkley was born in New York, in July 1842, and died in East Orange, N. J., November 6, 1911. During the Civil War he served in the Navy as assistant engineer, abandoning his studies at the College of the City of New York in order to enlist. At the close of the war he secured employment at the Morgan Iron Works as mechanical draftsman and constructor. Shortly afterwards he opened an office of his own in New York as constructing and consulting engineer and engaged in the manufacture of the Bulkley injector-condensor,

superheater, steam pumps, etc., continuing in this work until the time of his death. He was the inventor of the Bulkley injector-condensor, which he patented in 1875, as well as a special type of steam pump and improvements on superheaters.

Mr. Bulkley was a member of the American Institute of Mining Engineers and the American Institute of Electrical Engineers.

GEORGE B. CALDWELL

George B. Caldwell was born in Lowell, Mass., April 11, 1863. He received his early education in the public schools of Lowell and at the age of seventeen he became an apprentice of the Lawrence Manufacturing Company of Lowell, serving his time in their shops and drafting rooms. At the end of his apprenticeship he remained with the same company as assistant to the master mechanic until 1899, when he accepted the position of master mechanic of the Washington Mills Company, Lawrence, Mass., where for four years he was in charge of their drafting rooms as well as their construction and repair work. In 1893 he entered the employ of Westinghouse, Church, Kerr and Company and retained his connection with this firm until the time of his death, March 31, 1911. During this period he was in responsible charge of designing and constructing many works of engineering interest, among which may be mentioned the mechanical and electrical features of the South Terminal Station, Boston, Mass.; the Pittsburgh terminal of the P. & L. E. R. R.; the railroad shops of the P. & L. E. R. R. at McKees Rocks; the construction of the Kingsbridge Power Station of the Third Avenue R. R. He also supervised the design and installation of the Long Island Railroad Company's electrification, including the large power house at Long Island City. His last work was the design and construction of the mechanical and electrical features of the Pennsylvania Terminal Station in New York City, which he was obliged to relinquish just before it was completed. Mr. Caldwell was also a member of the American Society of Civil Engineers. He was a man of unswerving integrity and of great energy, inspiring perfect confidence and respect in all those with whom he dealt.

JOHN A. CALDWELL

John A. Caldwell was born August 12, 1849, at Johnstone, Scotland, and was educated at the Glasgow School of Design and Mechanical Engineering, from which he was graduated in 1867. He served

his apprenticeship as a pattern maker with the firm of Tweedale & Robinson, Johnstone, and as draftsman with Lawson & Son, Glasgow, from 1863 to 1868. In 1870 he came to the United States and was employed by Mackintosh, Hemphill & Company of Pittsburgh, for whom he designed large rolling mills and blast engines, and had entire charge of relining the Schoenberger blast furnaces and rebuilding the hot blasts at Pittsburgh, as well as the erection of two new blast furnaces at Port Washington, Ohio. He subsequently designed all the locomotives built by the National Locomotive Works, Connellsville, Pa., the Alice stamp mill and the Moulton mill at Butte, Mont., the elevator service for Z. C. M. I. at Salt Lake City, Utah, the stamp mill at Parrall, Mexico, the numerous filter plants for the Hyatt Pure Water Company, including the installation of the city plant at Oakland, Cal., and was in charge of the new engine and boiler installations for the H. W. Johns Manufacturing Company, Brooklyn, N. Y., contracted for and superintended the erection of numerous blower plants, water pumps and pumping stations for the P. H. & F. M. Roots Company, including the pumping station at Little Falls, N. J., and contracted for, designed and superintended the erection of numerous water-tube boiler plants, including piping, in conjunction with James Beggs & Company. In 1899 Mr. Caldwell became business manager for the American Stoker Company, New York, and in 1901 opened an office of his own in the same city, making a specialty of boiler room economies, particularly stokers and CO_2 recorders. He died on April 7, 1911.

WILLIAM LESTER CANNIFF

William Lester Canniff was born in Berea, Ohio, on May 23, 1862, and received his education in the public schools, acquiring his mechanical training through his own efforts and through experience. After obtaining a general knowledge as a machinist, electrician and stationary engineer, he decided upon tunnel work as his specialty and in 1896 entered the employ of the W. J. Gawne Company of Cleveland, contractors for the water works tunnel then in process of construction in that city, as master mechanic, and also assisted in the work of the second tunnel built shortly after the completion of the first. In 1900 he superintended the construction of the water works tunnel in Cincinnati, built by the same company, and had complete charge of the mechanical work. In 1905 he was employed by the Degnon Contracting Company of New York as mechanical superintendent

on the Belmont tunnel under the East River, and in 1907 by the United Engineering and Contracting Company on the construction of the Pennsylvania tunnel, New York City. At the time of his death on August 29 he was serving the T. A. Gillespie Company of High Falls, N. Y., in the same capacity, being engaged on the Rondout siphon tunnel of the New York City aqueduct. During the three years of his work for this company he designed and erected the largest compressed air plant ever built under one roof in this country. Although Mr. Canniff invented a number of appliances he patented only two, the Union hose and pipe coupling, and a pneumatic grout mixer extensively used through the country, on the subject of which he was a recognized authority.

JAMES CHRISTIE

James Christie was born near Ottawa, Canada, on August 28, 1840, and was of Scotch parentage. At the age of sixteen after a common school education, he came to the United States and under the guardianship of his uncle, one of the pioneer railroad constructors in this country, was employed with a railroad construction corps. In 1856 he served as an apprentice in the machine shop of locomotive works in Detroit, Michigan, and spent one year in Missouri as an assistant to engineers and contractors on the Pacific Railroad of Missouri. From there he went to Philadelphia, apprenticing himself to the I. P. Morris Company, proprietors of the Port Richmond Iron Works, where he learned the trade of a machinist. In 1865 he removed to Pittsburgh as superintendent and engineer of the Fulton Foundry, and later engaged in the designing and construction of iron works. In Phillipsburg, N. J., his next location, he devoted himself, as superintendent of the Phillipsburg Manufacturing Company, to the construction of iron bridges. In 1876 he became construction engineer with the Pencoyd Iron Works of A. and P. Roberts & Company, where he continued until after the works had been absorbed by the American Bridge Company. Here he did his most important engineering work. As general mechanical assistant and to a great extent his own draftsman, he began at once to improve the methods of work, introducing machinery and organizing the men employed, as a result making the works one of the largest and most efficient in the Eastern part of the country. Upon his retirement from the company he established a consulting practice of his own.

In 1884 Mr. Christie published Experiments on the Strength of

Wrought Iron Struts, based on his tests at Pencoyd, which won for him the Normal Medal given by the American Society of Civil Engineers, and made at other times numerous contributions to general and scientific literature. During the Civil War he served in the Antietam campaign and in 1863 entered the engineer corps. He was throughout his life interested in public affairs and held several political offices, serving as Mayor of Phillipsburg in 1870.

He joined the Society in 1885 and was one of its Vice-Presidents from 1902 to 1904. At the time of his death on August 24, 1911, he was serving his second term as President of the Engineers Club of Philadelphia, and was also a member of the Franklin Institute, the American Philosophical Society, the American Society of Civil Engineers, and Fellow of the American Association for the Advance-of Science.

WILLIAM HARRISON CORBETT

William Harrison Corbett, President of the Williamette Iron and Steel Works, Portland, Oregon, died at his home in that city on February 20, 1911, after a brief illness. Mr. Corbett was born in Brooklyn, N. Y., on October 31, 1868, and was graduated from Stevens Institute in the class of 1895. Before entering Stevens he had served his apprenticeship in the Rowland Machine Works, New Haven, Conn., and immediately upon graduation entered the field of mechanical engineering in New York City. In 1900 he removed to Portland, Oregon, to assume charge of the Williamette Iron and Steel Works, becoming President after the works were rebuilt some years ago.

Mr. Corbett followed the family trend as an iron manufacturer and ship builder, being the eldest son of Charles H. Corbett, Vice-President of the Continental Iron Works, and a grandson of the late Jeronemeus S. Underhill, one of the pioneer builders of iron vessels before the Civil War.

THOMAS B. DAVIS

Thomas B. Davis was born at Forest City, Ark., in 1876, and received his early education in the public schools and in the State University of Arkansas. After spending two years as draftsman on steel mill buildings, contracting plants and smelters in Denver, Colo., he studied for one year at Cornell University, securing upon its completion a position as designing engineer with the American Smelting

and Refining Company, New York, later going to Mexico as designer and superintendent of erection of the concentration mill at Santa Barbara, Chihuahua. Upon his return he accepted the chair of applied mathematics and machine design at the University of Nebraska, Lincoln, but after a year accepted a position as mechanical engineer with the Jeffrey Manufacturing Company, of Columbus, Ohio. While with this company he made estimates on and designed the cranes for handling the rock, sand, cement and concrete for building the locks at the Isthmus of Panama, manufactured by his company, and in December 1908, was asked by the Government to accompany the United States Engineering Corps to the Isthmus to look over the field and become familiar with the surroundings prior to the beginning of the steel work, one of the most gigantic undertakings in this line ever considered in this country. The success of Mr. Davis's work gave him a wide and enviable reputation for competency in his chosen field. At the time of his death, November 3, 1911, he was president of the Arkansas Farm Company of Little Rock.

JARVIS B. EDSON

Jarvis B. Edson died at his home in New York City, January 26, 1911. He was born in Janesville, Wis., April 30, 1845, and obtained his technical education at Cooper Union and at New York University, New York City. He saw active service during the Civil War, and being discharged from the army, July 1863, he connected himself with the South Brooklyn Steam Engine and Boiler Works, where he participated in the construction of several engines for United States war vessels notably the Mendota, Metacomet, Nyack and Nipsic. He later conducted a series of steam engine expansion experiments under the direct supervision of B. F. Isherwood, U. S. N., Hon.Mem Am. Soc. M. E., chief of the Bureau of Steam Engineering.

On November 1, 1864, he was appointed Acting Third Assistant Engineer in the United States Navy and at the close of the war received his honorable discharge. Returning to private life, he invented and perfected the Edson time and pressure recording steam engine. He also engaged in the manufacture of various instruments required by engineers, for the precision of which he was obliged to construct a mercurial column some 250 ft. high for high pressure alongside the Brooklyn towers of the East River Bridge. This instrument received due corrections for temperature and density, and differed from the guesswork method previously practised for obtain-

ing standards from which to lay off the dials of hydraulic and other high-pressure instruments. In 1873 Mr. Edson took part in the organization of the Domestic Telegraph Company and three years later devoted his attention to the manufacture and improvement of celluloid and zylonite. During this period he obtained many valuable patents on inventions, one of which was for a method of making artificial ivory in pyroxilin compounds. He also devised a novel method of sinking deep wells into clay, quicksand, gravel, etc.

Mr. Edson's naval experience, coupled with his characteristic patriotism, prompted his notable activity in organizing the New York Commandery, Naval Order of the United States, of which he was a charter member, and the Navy League of the United States. He was a member of the American Society of Naval Architects and Marine Engineers, American Society of Naval Engineers, Franklin Institute, National Geographic Society, Engineers' Club, Army and Navy Club, and several others.

HENRY A. FERGUSSON

Henry A. Fergusson was born in Philadelphia, Pa., December 1869, and received his technical training at the Spring Garden Polytechnic Institute and at Cornell University. In September 1887, he began an apprenticeship at the Baldwin Locomotive Works, Philadelphia, and from 1888 to 1902 was employed by the Pennsylvania Railroad Company at Altoona, Pa., holding successively the positions of assistant foreman of car shops; assistant master mechanic at the Meadows shops; assistant road foreman of engineers of the New York division, in sole charge of tonnage, rating and tests of locomotives; and assistant engineer of motive power. He resigned from the company to become assistant superintendent of motive power of the Chicago Great Western Railroad, St. Paul, Minn., and in 1904 entered the employ of J. T. Ryerson & Sons, Chicago, Ill., as engineer. At the time of his death on April 22, 1911, he had been consulting engineer and general manager for The Steel Roof Truss Company, Valley Park, Mo., for several years.

JOSEPH JAMES FERRIER

Joseph James Ferrier, whose death occurred at Fruitvale, Cal., October 29, 1911, was born at Brighton, Sussex, England, November 26, 1882. His education was obtained at the English common schools, and in 1904 he completed a course in civil engineering with

the International Correspondence Schools, during which time he was also employed by the Linlott Engineering Works, Horsham, England, in their foundry on municipal supplies and agricultural machinery. In 1903 he came to America and obtained a position with the Mergenthaler Linotype Company, Brooklyn, N. Y., as timekeeper, draftsman, estimator and designer. Two years later Mr. Ferrier secured a positiom in the electrification department of the New York Central & Hudson River Railroad Company, and rose from a subordinate place to chief draftsman in the steam engineering branch of this work. In April 1907, he entered the electrical engineer's office of the Southern Pacific Company, at San Francisco, Cal., taking a prominent position in the steam and mechanical engineering branches of the Oakland, Alameda and Berkeley electrification.

STUART E. FREEMAN

Stuart E. Freeman was born January 13, 1866, at Baltimore, Md He received his early education at the public schools and later attended the Franklin Institute in Philadelphia. From 1883 to 1887 he served an apprenticeship in the Illinois Central Railroad shops, Centralia, Ill., at the end of which period, he was employed by the Dickson Manufacturing Company, Scranton, Pa. He later accepted the position of foreman with the A. Falkenau Machine Company, Philadelphia. From October 1892 until May 1893 he was general foreman for the Philadelphia Hardware and Malleable Iron Works and until January 1895 for the Morgan Engineering Company, Alliance, O. Mr. Freeman was subsequently employed by the Todd-Stanley Mill Manufacturing Company, St. Louis, Mo., the International Drill Company, Barberton, O., the Stirling Tubular Boiler Company, the Fuller Manufacturing Company, New Haven, Conn., and at the time of his death he was general superintendent of the Smith-Furbush Machine Company, Philadelphia.

Mr. Freeman was a member of the Franklin Institute and of the Engineers' Club of St. Louis. He died in Norristown, Pa., February 2, 1911.

JOSEPH LEON GOBEILLE

Joseph Leon Gobeille, identified throughout most of his business career with the Gobeille Pattern Company, Cleveland, Ohio, was born in Poughkeepsie, N. Y., July 2, 1855. He obtained his mechanical training at Cooper Union, New York, and at the Rensselaer

Polytechnic Institute, Troy, N. Y. At eighteen years of age he entered the pattern shop of N. S. Vedder in Troy, leaving there five years later to take charge of the department of design and ornamentation with the Cleveland Stove Company, Cleveland, Ohio. In 1881 he established the firm of Gobeille & Brothers in the same city, retaining for himself the designing and drafting. In 1884 he bought out this concern and organized the Gobeille Pattern Works which he developed to large proportions. While he specialized in stove patterns, his shop turned out some of the largest and most intricate work called for by the leading foundries of the country. He invented and constructed ingenious and useful woodworking machinery and his pattern shop was provided with valuable and rare machine tools.

Leaving Cleveland in 1905 after a financial reverse, he was connected for a short time with the Abram Cox Stove Company, Philadelphia, Pa. Later he went to Niagara Falls, N. Y., and organized the Gobeille-Harris Pattern Company, of which he retained control until attacked by the illness which caused his death on September 27, 1911.

Mr. Gobeille was a past-president of the Cleveland Engineers Club and contributed papers at various times to meetings of foundrymen and stove manufacturers associations. He will also be remembered as a writer of short stories and a lover of rare books.

CHAS. ARTHUR HAGUE

Chas. Arthur Hague was born at Newton, Mass., October 9, 1849, and died June 26, 1911. He began his professional career in 1872 as a draftsman and designer with the Clapp and Jones Manufacturing Company, Hudson, N. Y., remaining in their employ until 1875 when he became mechanical engineer and draftsman on steam engines, boilers, etc., for the Frank Douglas Machinery Company, of Chicago. In the following year he resigned to enter the employ of the Furst & Bradley Manufacturing Company as a master mechanic. While there he patented important improvements in the plows and other implements manufactured by them and designed and constructed numerous special machines adapted to their line. In 1884 he became superintendent of the E. P. Allis Company of Milwaukee and three years later became connected with the Knowles Steam Pump Company of New York. The following year he entered the employ of H. R. Worthington, New York, as mechanical engineer, remaining there until 1895 when he established a consulting practice of his own.

Mr. Hague was the author of a book on Pumping Engines for Water Works.

ARNDT L. HAMMARBERG

Arndt L. Hammarberg was born October 4, 1862, in Gothenberg, Sweden, and received his technical education at the Chalmers Technical Institute, Gothenberg, from which he was graduated in 1881. His early experience was obtained with the Bergsund Shipbuilding Company, at Stockholm, and as designer and draftsman with the Donnarvest Iron and Steel Works. In 1889 he came to the United States to take a position with the Illinois Steel Company, remaining until 1891 when he entered the employ of the Homestead Steel Works. In the following year he was placed in charge of the rolling mill construction of the Morgan Construction Company, Worcester, Mass., and between the years 1895 to 1898 was successively connected with Julian Kennedy and with Mackintosh, Hemphill & Company, Pittsburgh, Pa., and with the Lloyd Booth Company of Youngstown, Ohio. In 1898 he entered the employ of the Wellman-Seaver Engineering Company, at Cleveland, as chief engineer, leaving there to become chief engineer in 1900 of the Ensley Steel Works, and in 1902 of the Youngstown Sheet and Tube Company. In 1907 he returned to Sweden, and until the time of his death on February 7, 1911, was employed as chief engineer at the Bergsund Shipbuilding Company at Falun.

FREDERICK L. HART

Frederick L. Hart died March 23, 1911, at Jamaica, N. Y. He was born July 3, 1861, in Brooklyn, N. Y., and was a graduate of the Polytechnic Institute of that city. He served his apprenticeship of a year and a half at the Columbian Iron Works of Brooklyn and three and a half years at the Wright Steam Engine Works of Newburgh, N. Y. His professional work included the erection of an ochre mill for the Franco-American Mining and Ochre Company of Brooklyn, and the installation and operation of an electric plant for the New York and Brooklyn Bridge. He was connected for a period of four years with the Third Avenue Railroad Company of New York as chief engineer and later in the same capacity with the Tenth Avenue and 125th Street Cable Railroads resigning later to become engineer in charge of the Broadway Railway Company. Here he remained for two years and then became general manager of the Baltimore City

Passenger Railway Company a position which he held for six years. Subsequent to this he was associated with the Washington Railway and Electric Company and the Thomas Bashor Company, contracting engineers of Baltimore.

CARYL D. HASKINS

Caryl D. Haskins was born at Waltham, Mass., May 22, 1867, and obtained his early education in the public schools of Philadelphia and Dedham, Pa. In 1879 he went to England, where he attended the Allison Towers and Roslyndale schools, devoted to the preparation of boys for military and engineering work, and later studied under the London University tuition system for a year. In 1887 he entered the employ of Haskins, Davis & Company, specializing in gas-engine work. Six months later he entered the shops of S. Z. Ferranti, where he was connected with the early work at the Old Grosvenor and Deptford stations. Returning to the United States in 1889, he was employed as designing draftsman by the Thomson Electric Welding Company, Lynn, Mass., but was soon transferred to the Thomson-Houston Electric Company as assistant foreman of the meter department. Shortly afterwards he was given charge of the company's outside meter interests. When the General Electric Company was formed, Mr. Haskins was put in charge of the meter interests of the consolidated company. In 1893 he was also given charge of the company's instrument department and in 1900 of the switchboard department.

While Mr. Haskins is best known as a meter expert, he was also an inventor in other lines, including the subjects of steel refining, meters, transformers, switches, circuit-breakers, time relays, etc. Perhaps his most important as well as most ingenious invention was an auto-dirigible torpedo, for which he refused several offers from foreign powers.

Mr. Haskins also found time for literary work. He was the author of a book on Transformers and of a work of fiction entitled, For the Queen in South Africa. He also lectured frequently at army and navy institutes. He had made a special study of electricity as applied to offense, and particularly to defense, in time of war and was considered an authority on this subject, and frequently acted in an advisory capacity to the Government in matters of this nature.

Mr. Haskins was a member of the American Institute of Electrical Engineers and the National Electric Light Association and was an

associate member of the Military Service Institute and the United States Naval Academy. He died November 18, 1911.

D. HOWARD HAYWOOD

D. Howard Haywood was born in London, England, March 26, 1869. He received a technical training in the Crystal Palace School of Practical Engineering during which time he spent several months in a wood-making pattern shop, machine shop and foundry. From June 1888 to May 1890 he was employed in the drafting office of Brewer and Company, engineers and contractors of New York City, and until January 1893 he was with Brown and Seward, patent attorneys. For the next seven years he conducted a business of his own in general engineering, mechanical and expert drafting work, in soliciting patents and in giving testimony as an expert in patent suits. At the time of his death, July 5, 1911, he was a member of the firm of Chapin and Haywood, New York City.

Mr. Haywood was a member of the New York Bar, New York County Lawyers' Association and foreign member of the Chartered Institute of Patent Agents.

ANDREW J. HEWLINGS

Andrew J. Hewlings was born in Philadelphia Pa., July 27, 1857, and received his early education in the public schools and Franklin Institute. In 1874 he entered the shops of L. Schutte and Company, where he served an apprenticeship of ten years. His later work included the erection of condensers, injectors, steam, water and air apparatus. He was president of the A. J. Hewlings Company, Chicago, Ill., until 1902, when he became manager of the catalogue department of the Crane Company, also of Chicago. This position he held at the time of his death on January 18, 1911.

LEMUEL R. HOPTON

Lemuel R. Hopton was born at West Stratford, Conn., on June 20, 1873, and received his early education in the public and high schools of New Haven and his technical training in the Sheffield Scientific School of Yale University, from which he was graduated with honors in 1896. Until 1900 he remained at Yale, teaching machine design in the department of Mechanical Engineering at Sheffield Scientific School, when he left there to enter the employ of Carl H. Schultz,

Inc., New York, as factory superintendent. In 1902 he resigned to take a similar position with the Enos Company of the same city, remaining with them until a short time before his death on September 5, 1911.

Mr. Hopton was the inventor of several electrical appliances used by the Enos Company, and also of the Opalux glass, used for high candle power lighting and manufactured by the Opalux Company, of which he was an officer and director. He was the author of many articles published in the electrical magazines and a member of the Illuminating Engineering Society, before which a number of his papers were presented.

CHARLES SCRANTON HUMPHREY

Charles Scranton Humphrey, who died in an automobile accident on June 29, 1911, was born at Kokomo, Ind., September 2, 1888. Two years later he moved with his parents to West New Brighton, N. Y. He received his preparatory education at the Westerleigh Collegiate Institute and entered Cornell University in 1904, receiving the degrees of B.A. in 1908 and M.E. in 1910. He was interested in automobile construction and also spent several months in the gas engine department of the Bethlehem Steel Company, South Bethlehem, Pa., and later in the shops of the C. W. Hunt Company, West New Brighton, assisting in the design and construction of a combination fuel oil electric locomotive for use on the west coast of South America. He wrote a number of papers on automobile design and construction and had for some time been a regular contributor to one of the leading journals devoted to the motor industry.

CHARLES WALLACE HUNT

Charles Wallace Hunt, Past-President of The American Society of Mechanical Engineers and of the United Engineering Society, was called to his rest on March 27, 1911, filling to the full his allotted three score and ten years.

He was born at Candor, Tioga County, New York, on October 13, 1841. His early scientific education was begun at Cortland Academy, Homer, New York, but during his life he never ceased to be a student. During the Civil War, Secretary Stanton assigned to him the duty, for which his kindly nature and sound judgment admirably fitted him, of looking after the welfare of the thousands of negroes who flocked into the Union lines from the Southern States. He taught

them how to take care of themselves by putting them at work in the mills furnishing supplies for the Government.

Returning from his services to the Government, about 1868, he began his business career on Staten Island, New York, in the storing and handling of coal. It was here that his engineering career began. He realized how slow and expensive the process of handling coal then was and set about to improve it. The result was the development of his automatic railroad, by which the coal was lifted from barges and carried back on an elevated track and dumped automatically at any desired point. This system was later introduced to a large extent, notably at the large coal stations of the Calumet and Hecla mines on Lake Superior, and in many other lake and sea ports in this country and abroad. The automatic railway completely solved the problem, and so perfectly that few if any improvements in this particular device have been made since. This work led to the invention of many coal-handling devices.

Later he became interested in the development of the system of chain conveyors running on wheels and having swinging buckets which always remained upright except at the points where they were dumped. This system was very largely and successfully used in many large steam power plants and locomotive coaling stations. By its use coal could be carried horizontally, or vertically, or at any angle, from any point and delivered automatically and continously at any one of many other points; and ashes could be collected at many different points and delivered at other points by the same endless conveyor chain with buckets carried on wheels, and kept continuously in motion by an ingenious pushing device that avoided very largely the friction of the ordinary sprocket chain machines, then in common use, which dragged the coal along the bottom of a trough.

A story is told of him by one of his friendly competitors which is both characteristic of the man and vividly brings out the difference in principle of these two systems. The two men were before a committee which had to decide on the purchase of one of the two systems presented. Mr. Hunt said: "If you wished to move a cat from one corner of the room to the other with the least effort and trouble, would you catch it by the tail and drag it across against the resistance of its claws in the carpet, or would you gently lift the cat into a basket and smoothly and easily carry it across?" He got the contract and his competitor enjoyed the story.

His genial humor, coupled with his straightforward common sense and ability to get at foundation principles, was among his most attractive characteristics.

In connection with his coal-handling machinery, Mr. Hunt was called upon to design and build many large steel structures for the storage of coal in large quantities. These included the coal stations for the United States Navy at Guantanamo, Puget Sound and Manila, as well as other large plants in South Africa, Europe and Australia.

In addition to the large plant of the C. W. Hunt Company on Staten Island, New York, of which he was president, there were plants in England and Germany for the manufacture of machinery designed by him. The development of coal-handling machinery by Mr. Hunt and his associates has reduced the cost of handling coal from about thirty to three cents per ton. He designed machinery for the rapid and economical handling of freight, a notable example being that at the Greenville terminal of the Pennsylvania Railroad.

Another direction in which Mr. Hunt showed his ingenuity and good engineering sense, was in the development of his industrial railway. He adopted a narrow gage of $21\frac{1}{2}$ inches for his tracks, which were made in complete sections ready to be joined together in any desired combination with very short curves and switches and turntables enabling the tracks to be carried anywhere around shops and yards. His cars were made in a great variety of special forms to do special work in the best manner. The wheels of his cars have their flanges on the outside, contrary to the common practice, for which he claimed great advantage in reducing the resistance of the car on the track and particularly on short curves. He was probably the first to build and use electric storage battery locomotives on very narrow gage railways having short curves. These locomotives involved special and unusual problems which he showed great ingenuity and skill in solving.

His inventive genius is represented by one hundred and forty-seven patents issued to him and three others which have since been allowed.

Mr. Hunt had a deep interest in the profession of engineering and was always ready to devote his time and energy to its advancement. His private interests were always held secondary to his interests in the various engineering societies of which he was a member.

He was a member of the Society for 26 years and was elected one of its Vice-Presidents in 1892 and became its President in 1898. He has served on many of its most important committees and his devotion to the work has been manifest in all the large policies adopted by the Society. His most notable committee work was done as Chairman of the Committee appointed by the Council in 1902 to draft a new Constitution and By-Laws. The result of the assiduous work of that Committee, of which

he bore the major part, was adopted by the Society in 1904 and then became its organic law. He proposed the policy, incorporated in the Constitution, of putting the various activities of the Society in charge of Standing Committees of five members each, one member retiring and one new member coming on each year, by which a large number of men, devoted to the best interests of the Society have been brought into its active management. He remained Chairman of the Committee on Constitution and By-Laws to the time of his death, and his last work for the Society was done in that Committee on February 17 last. In no place will his broad-minded yet conservative counsel be more greatly missed.

Mr. Hunt ceased to be a statutory member of the Council in 1904, but he, with other Past-Presidents, was invited by the Council to sit at its sessions as Honorary Councilor. Although not obliged to be present, he rarely missed a Council meeting when his health permitted and always gave such sound advice that the members of the Council were wont to show their sincere respect by referring to him as our "Elder Councilor."

Mr. Hunt was also a member of the American Institute of Electrical Engineers; the American Institute of Mining Engineers; the American Society for the Advancement of Science; Franklin Institute; New York Electrical Society; Inventor's Guild; New York Chamber of Commerce; National Geographic Society; Staten Island Association of Arts and Sciences; Staten Island Chamber of Commerce; The Engineers' Club; The Railway Club; The Hardware Club; and The Machinery Club. He was President of the Richmond County Savings Bank.

When Mr. Andrew Carnegie made his notable deed of gift in 1903 to be devoted to a building for the Engineering Societies, Mr. Hunt was promptly chosen as a member to represent The American Society of Mechanical Engineers on the organization committee, and later of the building committee, and finally of the Board of Trustees of the United Engineering Societies. He continued in office during two terms under the principle, which he himself recommended, that no trustee should be immediately eligible at the close of a second term. After a year had elapsed however, he was re-elected to serve in the position which he had so conspicuously adorned. He was at the beginning of his third term when death overtook him. His influence in the Board was conservative, yet progressive. It was his recommendation that no decision of the Building Committee should ever be carried out until the entire Board was unanimous respecting it. This was

largely responsible for many well-balanced decisions respecting details of the building and its consequent satisfaction to all parties. Up to the very beginning of his last illness he was active and suggestive respecting certain propositions looking to the development of the library and an enlargement of its scope and usefulness.

Mr. Hunt served also on many other committees representative of the Society. About a dozen years ago he served as a representative of the Society in formulating a revision of the Building Code of the City of New York to take care of the changes introduced by steel construction and the tall cellular building. He served also on the committee to install the Thurston memorial tablet and was a member of the John Fritz Medal Board. He also served on a committee to write the history of the Society. From time to time he contributed from his experience several valuable papers to the Society.

In his presidential address at the Annual Meeting in December 1898, Mr. Hunt took as his title, The Engineer, and in that paper, drawing most largely perhaps upon his own philosophy of life, he made this statement, doubtless also the product of his own experience: "The life of an engineer has a full measure of the labors, the trials, the discomforts and the disappointments which are found in this as in every other walk of life, but—it has also the successes which come from well directed labors. It is not, however, either the useful work in itself, or what are called the successes of life, which bring happiness. It is a man's ideals which make him happy. . . . . Cheerfully can he enter upon his daily task with the consciousness that his application of these discoveries is of real service in lightening the burdens of our life, as well as elevating and ennobling his fellow men." Such was the story of Mr. Hunt's life and such were his achievements.

Col. E. D. Meier, President of the Society; Prof. F. R. Hutton, Past-President and Honorary Secretary; Henry R. Towne, Past-President; Jesse M. Smith, Past-President; Alex. C. Humphreys, Vice-President; Henry G. Stott, Manager; Charles Kirchhoff, John W. Lieb, Jr., Alfred Noble, W. H. Fletcher, Members; and Calvin W. Rice, Secretary, were appointed Honorary Vice-Presidents to attend the funeral and represent The American Society of Mechanical Engineers. This was held in the Church of the Ascension at West New Brighton, on Thursday, March 30. The body of Mr. Hunt was laid at rest in the beautiful cemetery at New Dorp, Staten Island, New York.

HENRY JAMES JOHNSON

Henry James Johnson was born in Providence, R. I., July 7, 1842. He was educated in the Providence public schools and in 1861 was graduated from Brown University with the degree of C.E. From 1863 to 1891 he was employed by the Providence Steam Engine Company, first as draftsman and later as designing engineer. He designed and superintended the construction of the Nagle pumping engine, the Burdict bolt and nut forging machines, steam capstans for the United States Navy, the improved Greene automatic drop cut-off engine, stationary and marine boilers, and others, upon several of which he secured patents. In 1891 he became designing engineer for Plumb, Burdict and Barnard, Buffalo, N. Y., bolt manufacturers. Two years later he again became connected with the Providence Steam Engine Company in the capacity of consulting engineer, and from 1898 to the date of his death on October 8, 1911, was consulting engineer for Filer and Stowell Engineering Company's eastern office.

WARREN S. JOHNSON

Warren S. Johnson was born in Leicester, Vt., November 6, 1847, and died December 5, 1911. At an early age he moved with his family to Waukesha, Wis., where he attended school and learned the printer's trade. At seventeen years of age he became foreman in a newspaper office at Durand, Wis., at the same time attending high school. He taught school in Menominee for a number of years and then became county superintendent of schools. In 1876 he was appointed professor of mathematics, sciences and drawing at the state normal school at Whitewater, Wis. Here he maintained a private laboratory where he experimented with storage batteries and where he conceived the idea of automatic temperature regulation. In 1883 he resigned his professorship and formed a partnership with William Plankinton of Milwaukee, where he developed the Johnson system of temperature regulation, now used all over the world. In 1885 the partnership became a corporation known as the Johnson Electric Service Company, the name being changed later to the Johnson Service Company.

The temperature regulator was not Mr. Johnson's only invention. Others were a brass foundry furnace, gasolene gas machine, pneumatic clock and time system, wireless telegraph system, automobile parts and various devices used in connection with the business in which he was engaged. The huge pneumatic clock at the St. Louis

Exposition in 1904 was made by Mr. Johnson. He also constructed the city hall clock in Philadelphia, the largest in the world, and the city hall clock in Milwaukee which has the largest bell in the world.

Mr. Johnson was a member of the Franklin Institute, the American Institute of Electrical Engineers and the American Society of Heating and Ventilating Engineers.

CHARLES J. LARSON

Charles J. Larson was born on March 2, 1872, at River Falls, Wis. His early education was obtained at the State Normal School, Morehead, Minn., and at Macalester College, St. Paul, Minn., and his technical training at the Rose Polytechnic Institute, Terre Haute, Ind., from which he was graduated in 1900 with the degree of B.S., and from which he also received the degree of M.E. in 1909. In 1900 he entered the employ of the Allis-Chalmers Company at Milwaukee, as erecting engineer, and from 1905 to 1907 had supervision of the installation, operation and testing of machinery furnished by this company in the Eastern territory. Among the prominent engineering works which he superintended in this capacity were the Washington drainage system and water works, the Boston sewerage system, the Boston elevated railway, the New York subway engines, the New York high-pressure service, and the Midvale Steel Company's plant and machinery for the Connecticut Railway Company. In 1904 he installed and had charge of the operation of the 6000-h.p. Allis-Chalmers-Bullock exhibit at the St. Louis Exhibition.

He was forced to resign his work in 1908, because of ill-health and accepted a position as chief engineer of the Union Electric Company, Dubuque, Iowa, which he held at the time of his death on April 6, 1911.

Mr. Larson was a member of the American Institute of Electrical Engineers and the National Electric Light Association, and had made many valuable contributions to technical journals.

WILLIAM S. McKINNEY

William S. McKinney was born in Troy, N. Y., August 11, 1844. He removed with his parents to Cincinnati, Ohio, in 1861, where his father engaged in the manufacture of hardware in partnership with Miles Greenwood. At the age of twenty, after the sudden death of his father, he assumed entire charge of the factory, continuing to carry on the business until the expiration of the partnership

agreement with Mr. Greenwood. He then, together with his brother J. P. McKinney, built a small works for the manufacture of hardware, making a specialty of strap and tee hinges and butts. Recognizing the advantages to be found in Pittsburgh for a business of this character, they removed the works to Allegheny, Pa., and organized the McKinney Manufacturing Company, of which he was president until his death. His engineering work was confined mainly to the designing, building and improving of machinery adapted for the manufacture of heavy hardware. He died at his home in Pittsburgh, August 30, 1911.

JAMES McLAUGHLIN

James McLaughlin who died on August 18 at Clifton Springs, N. Y., was born on May 8, 1867 at Castlefin, County Donegal, Ireland, coming to the United States with his parents at the age of fifteen. Here he entered the Philadelphia High School and later the University of Pennsylvania, where he received his training as a civil engineer. In 1885 he entered the employ of the Philadelphia & Reading Railway, as secretary to the general counsel, and two years later became bookkeeper and correspondent for the Philadelphia Engineering Works. Shortly afterward he decided to take up the practical and technical side of mechanical engineering and entered the shop of the company, also taking a course of lectures on mechanics at the University of Pennsylvania. In 1889 he became general manager of the Barr Pumping Engine Company of Philadelphia, and designed and installed many large high-duty pumping engines, principally in New England but also in Kansas City, Harrisburg, Trenton, Denver and other cities.

In 1903 he resigned to establish the firm of McLaughlin Brothers, Incorporated, for the design, construction and equipment of buildings, including reinforced concrete, steel and New England mill construction, power plants and electrical equipment, and plumbing and sprinkling systems. In the following year the main offices were removed from Philadelphia to Baltimore and many of the large buildings of Baltimore were erected under Mr. McLaughlin's supervision. He was at the time of his death president and general manager as well as senior member of the firm.

WILLIAM B. MASON

William B. Mason was born at Durham, Me., December 22, 1852, and died at his home, Dorchester, Mass., February 4, 1911. He received a common school education and first went to work in a machine shop at Biddeford, Me. He later came to Boston and was employed at the Hinckley Locomotive Works and as engineer on various harbor steamers. Mr. Mason then joined the navy as machinist and was assigned to the U. S. S. Omaha. He served for two years on the Pacific Station off the west coast of South America. Returning to Boston he accepted a position with Cressey and Noyes and while there invented the Mason steam pump speed governor, which is today the standard method of control of direct-acting steam pumps. In 1882 the Mason Regulator Company was organized to manufacture the instrument. When this enterprise was well under way Mr. Mason turned his attention to the design of a pump pressure regulator for controlling the discharge pressure on steam driven pumps. He later adapted the same principle to a reducing valve of which many thousands have been made for use under a great variety of conditions. When steam heating was adopted for use on railroad cars, the Mason reducing valve was found to be the only one that could be satisfactorily employed and the leading railroad systems of this country, Great Britain and continental Europe have made it their standard. The business was broadened by numerous other devices for controlling steam, water and air at all pressures. When the manufacture of steam automobiles was first attempted by the Stanley Motor Carriage Company, Mr. Mason designed their engine and afterwards manufactured several thousand for the various steam carriage manufacturers.

He was a member of the New England Railroad club.

EDGAR W. MIX

Edgar W. Mix was born at Columbus, Ohio, February 4, 1867, and in 1888 was graduated from the Ohio State University with the degree of B.S. The two years following were spent in the employ of the Thomson-Houston Electric Company, Lynn, Mass., in general shop work and practice and in the development of the Thomson recording watt meter. In 1890 he took this meter to Paris where it was awarded first prize in the meter contest. The following five years were devoted to general engineering, designing and installing various electric railway and lighting plants in France and Russia.

In 1895 he became chief engineer of the Société des Etablissements Postel-Vinay, Paris, which concern was then controlled and has since been absorbed by the Thomson-Houston Electric Company. In this capacity Mr. Mix played an important part in the development of all the electrical machinery which the company have been building since he joined them, as well as steam turbines and modern extraction machinery for collieries. He left the Thomson-Houston Company in October 1911, to take up the European branch of the General Motors Export Company of America.

Mr. Mix was a member of both the Aero Club of France and the Aero Club of America, and represented both countries in the international balloon race from Zurich on October 5, 1909, in which he was the only American contestant. He died suddenly November 12, 1911.

DOUGLASS G. MOORE

Douglass G. Moore was born at Corning, N. Y., February 26, 1846, and died at Norfolk, Va., March 7, 1911. He was educated in the public schools of Elizabeth, N. J., and in Dr. Foote's Academy, and served his apprenticeship under his father, Samuel L. Moore, of Samuel L. Moore and Company. From 1866 to 1870 he engaged in marine practice, returning in the following year to work with his father in the large machine shop and foundry of the company at Elizabethport, N. J. He sold his interest in this concern, however, to the American Shipbuilding Company when the latter firm was formed, and purchased a controlling interest in the Port Johnston Towing Company and the Poughkeepsie Brass Company, with both of which he retained active connection until his death. Mr. Moore was greatly interested in public affairs and was ever eager to adopt useful improvements.

CHARLES HILL MORGAN

Charles Hill Morgan was born January 8, 1831, in Rochester, N. Y. His parents, Hiram and Clarissa L. (Rich) Morgan, were of old New England stock, the line of his father going back to Miles Morgan, one of the founders of Springfield, Mass., who came to this country in 1636, from Bristol, England.

Mr. Morgan's father having been a mechanic of limited means, the son Charles was obliged to work in a factory at the age of twelve, and his early education was that afforded by the Massachusetts district school of seventy years ago, and short terms in the Lancaster

Academy. When fifteen he entered the machine shop of his uncle, J B. Parker of Clinton, Mass., as an apprentice.

At seventeen he determined to learn mechanical drawing and through his efforts a class for the study of this subject was formed. taught by John C. Hoadley, late member of The American Society Mechanical Engineers, then civil engineer of the Clinton Mills. Those few lessons in drawing, taken at night, after twelve hours of work in the shop, were the most important factor in establishing Mr. Morgan's mechanical career, and perhaps of several others in that class.

In 1852, when twenty-one, Mr. Morgan was put in charge of the Clinton Mills dye-house. He devoted himself to the study of chemistry with great zeal, and filled his new position with entire success, gaining valuable experience in the management of subordinates.

For a time Mr. Morgan was draftsman for the Lawrence Machine Company. Later, from 1855 to 1860, he was mechanical draftsman for the distinguished inventor and manufacturer, Erastus B. Bigelow. In association with him and Charles H. Waters, the agent of the Clinton Wire-Cloth Mills, Mr. Morgan gained an invaluable experience and may be said to have been trained in a hive of invention. Mr. Morgan introduced a system of designing and constructing cam curves for looms. This system proved of great value and was later the subject of a valuable paper read before the Worcester Polytechnic Institute, and subsequently published by Mr. Morgan in pamphlet form.

Forming a partnership in 1860 with his brother Francis H. Morgan, he was for several years engaged in the manufacture of paper bags in Philadelphia, and during a part of this time a paper mill was operated by the firm near Coatesville, Pa. Previous to this time the imperfections in machinery had made paper bag making in the United States anything but a success. Mr. Morgan perfected the equipment so that the business was placed on a commercial footing

In 1864 Hon. Ichabod Washburn was in need of a superintendent for his works, for the manufacture of wire, at Worcester, Mass. His friends at Clinton, engaged in the manufacture of machinery and wire-cloth, warmly recommended Mr. Morgan. Mr. Washburn accordingly engaged Mr. Morgan as superintendent of manufacturing for the firm of I. Washburn and Moen of Worcester, Mass. Four years later, when a joint-stock company was organized and incorporated under the name of Washburn and Moen Manufacturing Company, Mr. Morgan became general superintendent. He made many trips to Europe for the purpose of visiting the mills of England, Belgium, Germany, France and Sweden. Through these visits,

through the publications devoted to wire manufacturing, and through patents issued both in Europe and America, he kept himself informed of all changes made or improvements adopted. The fruit of this devotion was seen in the increased excellence, variety and amount of the company's manufactures. He was for eleven years one of the directors of the company.

Mr. Morgan has been most prominently identified with the development of the continuous rolling mill, and today in steel centers the world over the continuous mill is known as the Morgan mill.

The first continuous mill was designed and originally constructed by Mr. George Bedson, in Manchester, England. One of these mills was purchased by Washburn and Moen Manufacturing Company, and erected in Worcester in 1869, and constituted a great advance over the rolling previously practiced. It soon became evident that the means of handling the product of the mill were inadequate and the first important step in development was the power reel designed by Mr. Morgan to replace the old-time hand-operated reel.

Mr. Morgan's second, and very important contribution to this system, that, indeed, which marked the great difference between the Bedson mill and the Morgan mill, was the practical development of a continuous train of horizontal rolls, the Bedson mill having had alternate sets of horizontal vertical rolls. This was accomplished by providing intermediate "twist guides," which gave to the metal being rolled the necessary quarter turn between the successive sets of rolls, and proved to be so successful that the Morgan mill is the only type of continuous mill now in use. Nine years after the contruction of the Bedson mill, another mill, from new designs furnished by Mr. Morgan, was built on the Belgian and continuous plans. This mill, the result of Mr. Morgan's studies, was known as the combination mill.

The third improvement was the invention, by Mr. Morgan, of automatic reels, both of the pouring and laying types, such as are now in common use in every rod mill in the world. These reels were completed and a successful test made March 10, 1886.

In 1887, after twenty-three very active years as General Superintendent of Washburn and Moen Manufacturing Company, declining health led Mr. Morgan to resign his position, and take up what then seemed likely to prove less arduous cares.

Some years, before in 1881, he had founded the Morgan Spring Company, for the manufacture of springs, and was thus the pioneer in this line of business which, at the present, is engaged in by many firms.

Manufacturers of steel products, at this time sought his advice in engineering problems, and his reputation as an engineer was in this way widely extended. This work led directly to the formation, in 1891, of Morgan Construction Company, of Worcester, manufacturers of rolling mill and wire drawing machinery.

Mr. Morgan was active in religious, corporate, charitable and educational lines throughout his life. He was closely identified with the growth and success of the Worcester Polytechnic Institute, having been a member of its Board of Trustees since the Institution was founded about forty-five years ago. In this capacity he has rendered a service of signal importance, greater even probably in its far reaching effect than his achievements in the profession of engineering. When in March 1866, the Hon. Ichabod Washburn made his gift to establish the machine-shop and working mechanical department of the Worcester Polytechnic Institute, Mr. Morgan was elected by the trustees as one of their associates, with the expectation that he would give the shop the benefit of his great mechanical genius and large experience. The expectation was not disappointed. Mr. Morgan's sagacity, his constant oversight, his inventive genius and his great business capacity were constantly at the service of the school, and the machine shop has been entirely successful, recognized everywhere as a most important and valuable part of the Institute.

In 1893, Mr. Morgan served on the Board of Judges of the World's Exposition in Chicago.

In 1900 when the Navy Department was embarrassed by conflicting commercial and political claims in its award for special gun lathes, Secretary Folger nominated Mr. Morgan one of the committee of three distinguished engineers to pass upon the merits of so vital a question.

Although many years a member of The American Society of Mechanical Engineers, he was very much surprised, when in Europe in 1899, to receive a cablegram announcing his nomination to the wholly unsought office of President. In the following December he was unanimously elected to that office and served for one year. This year, 1900, was a notable one in the life of the Society, for a joint meeting was held in England with The Institution of Mechanical Engineers. On this trip three sovereigns, Victoria of England, Oscar of Sweden and Leopold II of Belgium, summoned Mr. Morgan to an audience. In France the unusual distinction of election to honorary membership in La Société des Ingenieurs Civils de France was conferred upon him.

Mr. Morgan was always an admirer of Henry Cort, the inventor of the art of puddling iron with coal and of rolling metals in grooved rolls, and in his Presidential address before the Society, Some Landmarks in the History of the Rolling Mill, he paid a tribute to the Englishman's genius. In 1905 he had two bronze tablets erected in memory of Cort, one of which he presented to the church at Lancaster, Cort's birthplace, and the other to the church at Hampstead, where he was buried. The tablets which are exactly alike contain a finely executed bust of Cort with an appropriate inscription.

At the time of his death, January 10, 1911, Mr. Morgan was President of the Morgan Spring Company and the Morgan Construction Company. He was a member of the American Institute of Mining Engineers, the British Iron and Steel Institute, The Institution of Mechanical Engineers of Great Britain, the Engineers' Club of New York, and an honorary member of the Society of Civil Engineers of France. He was also one of the founders of Plymouth Church and its first Sunday School Superintendent and for many years served on its board of deacons.

Chas. M. Allen, George I. Alden, F. H. Daniels, J. P. Higgins, Geo. I. Rockwood and C. J. H. Woodbury were appointed Honorary Vice-Presidents to represent the Society at the funeral of Mr. Morgan.

HUGH S. MORRISON

Hugh S. Morrison was born in Petersburg, Va., July 7, 1877, and died at Saranac Lake, N. Y., August 3, 1911. He was educated in the public school and at the Virginia Mechanics Institute, Richmond, where for two years he was a student. He obtained his shop experience at the Richmond Iron Works, where he was foreman of the boiler shop, and at the Petersburg Iron Works, Roanoke Rapids, N. C., being in charge of the outside construction work. In July 1898 he entered the drawing-room of the Richmond Locomotive Works and later became assistant to the superintendent in charge of the layout and construction of the new Smith boiler, machine and special shops and track system in the yard. He opened an office of his own in 1901, and was for a short time associated with Mr. Kinderwater. During this period he designed the machinery and layout and the heating and ventilating systems for the Baughman Stationery Company, as well as the layout of the entire building. In 1903 he became a member of the firm of Myers, Finney & Morrison, designing

and constructing special machinery, and was responsible for the drawings and specifications for the Keely cotton compress for that company, Chester, S. C. Mr. Morrison organized the Morrison Machinery & Supply Company of Richmond, Va., in 1905, with which he was actively connected until his death.

JOHN O. NORBOM

John O. Norbom was born at Fredrikssbad, Norway, September 12, 1865. He was graduated from the Horton Technical School, Horton navy yard, Norway, in 1883 with the degree of M.E., and for two years worked as draftsman at Fredrikssbad Mekaniske Verksted, Norway. Mr. Norbom then came to this country and was employed at the Risdon Iron Works, San Francisco, Cal., as engineer in the mining department. In 1895 he accepted a similar position with the J. Hendy Iron Works and two years later became manager of the mining department of the British Columbia Iron Works, Vancouver, B. C. He next held the position of mechanical engineer in the mining department of the Union Iron Works, San Francisco. From 1901 to 1903 he was consulting engineer at the East Rand Proprietary Mines, Johannesburg, Transvaal. After the Boer War he returned to his native country, where he examined mines for a London mining syndicate. In 1908 he resumed his work in California as mining engineer.

Mr. Norbom met his death by an accidental explosion on a ferryboat on the Bay of San Francisco, January 13, 1911. He was a member of the American Institute of Mining Engineers and the Polytechnical Society of Norway.

JOHN G. OULD

John G. Ould died at his home in Brooklyn, N. Y., on December 21, 1911. He was born April 2, 1863, in Falmouth, England, and was educated in the private schools in England and in the Kensington Science and Art School. He also took courses in Pratt Institute and in the Polytechnic Institute of Brooklyn, N. Y. During his youth he was an apprentice in the English shops of A. & W. Robertson, Victoria Docks, London, builders of marine engines; and Waddington and Company, Liverpool. In May 1885 he took the Board of Trade examination for second class engineer, and for the following five years spent most of his time at sea. During this period he was chief engineer of the steamship Montana, sailing between England

and Buenos Aires; chief engineer of the steamship Condor, covering the same route; and chief engineer of the steamship Euxine, from England to Constantinople. In June 1890 Mr. Ould came to New York and obtained employment with Seabury & Company and later with the firm of Mitchell and Boyeson as foreman in charge of repairs. He was subsequently engaged as assistant engineer with the Western Union Company and with the Ball & Wood Company. In February 1897 he superintended the installation of the mechanical plant of the Polhemus Memorial Clinic and held the position of superintendent and chief engineer of that institution until the time of his death.

THOMAS F. SALTER

Thomas F. Salter was born in Prattsville, N. Y., March 31, 1875, and after his graduation from the Syracuse, N. Y., high school, learned his trade in the machine shops of the Syracuse Specialty & Manufacturing Company, and through correspondence schools and private tutors. The several years following he was employed by the General Electric Company, Schenectady, N. Y., and by the Western Electric Company, New York City. From 1902 to 1906 he was assistant to the executive engineer of the C. W. Hunt Company, West New Brighton, N. Y., being instrumental in designing machinery for a number of coal-hoisting installations for the United States Government, various railroads, and other users of such apparatus, and was considered by them an engineer and designer of unusual ability. He then entered the employ of the North Penn Iron Company, Philadelphia, Pa., as chief engineer and manager of the shop. While here he designed and constructed a special 20-ton hand-traveling crane for the Cambria Steel Company, intended for operation in a building of semicircular form. He was also responsible for three 5-ton 3-motor electric traveling cranes for the Bureau of Supplies and Accounts, Navy Department, which were erected at the Charleston Navy Yard. At the time of his death, October 25, 1911, he was manager of sales and engineering with the Standard Roller Bearing Company, Philadelphia, Pa.

Mr. Salter was a member of the New York Railroad Club, the Society of Automobile Engineers, the Interurban and Street Railway Association and the Sales Managers Association.

EDWARD K. SANCTON

Edward K. Sancton was born in St. John, New Brunswick, June 11, 1840, and served his apprenticeship with the Fleming Foundry, machinists and engineers. In 1861 he left St. John to enter the employ of the Morgan Iron Works in New York City and was soon placed in charge of the construction and erection of the engines and machinery for the double-turreted monitor, Onondago, on the completion of this work becoming foreman of one of the departments. He later became engineer on the steamship Fulton, plying between New York and Southampton, which was the pioneer in the attempt to establish a trans-atlantic line under the American flag subsequent to the Civil War. When the steamship company failed to maintain a profitable transatlantic service, he returned to shop work and was employed by Fletcher and Harrison, New York. During this period George W. Quintard, former proprietor of the Morgan Iron Works, sold this establishment and organized a new marine engine works, called the Quintard Iron Works, of which Mr. Sancton was made foreman of the machine department and assistant to the manager. In this position he supervised the construction and erection of many large pumping and marine engines. In the year 1881 he entered the employ of E. D. Leavitt, Jr., consulting and designing engineer, Cambridge, Mass. This engagement was fruitful of an extended experience, the inspection covering work constructing with the I. P. Morris Company, in Philadelphia, the Dickson Manufacturing Company, Scranton, Pa., and others, resulting in the reorganization of the Dickson Company in 1883 and in the appointment of Mr. Leavitt as consulting engineer, with Mr. Sancton as assistant superintendent and works manager. He continued in this position until 1895 when he was called by Fraser and Chalmers, Limited, of Erith and London, England, to assume the position of general manager of their English works. In 1910 he was appointed a director of this company and took an active interest in their affairs until his death on December 26, 1911.

FRANCIS SCHUMANN

Francis Schumann was born in Thuringia, Saxony, in 1844, and came with his family to America during the disturbances in 1848. Here he became interested in engineering and served throughout the Civil War as a topographical engineer on General Meade's staff, becoming engineer to the Treasury Department at Washington at its close. Eventually he resigned to become president of the Phoenix

Iron Works of Trenton, N. J., leaving there in 1887 to found the Tacony Iron and Metal Company at Tacony, Pa. During his association with the company, as its president and general manager, the tower of the Philadelphia City Hall, including the William Penn statue and the smaller bronze groups, many of the United States coastwise lights, and the ornamental bronze work of the Congressional Library at Washington, were constructed. He later became president of the Pennsylvania Iron Works Company.

Mr. Schumann was a past-president of the Engineers Club of Philadelphia, the first president of the American Foundrymen's Association, and a member of the Franklin Institute and of the American Institute of Architects. He was instrumental in founding the Tacony and the Pelham Trust companies and was vice-president of each. He died on June 29, 1911, at his home in Germantown, after a prolonged illness.

JOHN WRIGHT SEAVER

John Wright Seaver was born in Madison, Wis., January 8, 1855, and died in Cleveland, O., January 14, 1911. He was educated in the public schools of Buffalo, N. Y., and at thirteen years of age entered the machine shop of the Shepard Iron Works of that city. His technical education was obtained at a night school to which he was obliged to walk three miles from his home, after working hard all day.

In 1873 he entered the machine shop of the Howard Iron Works, and at the age of twenty he was employed by the Buffalo Car Company as assistant superintendent in charge of about one thousand men. Leaving there, he formed the firm of Seaver and Kellogg Company, where he designed and built the first steel cars in this country. These were not commercially successful as they were too far in advance of their time. After spending a year and a half in this business, he became assistant engineer of the Kellogg Bridge Works, where he made structural iron his specialty. He began at this time to attract outside attention as an engineer. In 1880 Mr. Seaver moved to Pittsburgh to become chief engineer for the Iron City Bridge Works, and while in their employ designed and built a number of notable bridges and other steel structures. In 1884 he became chief engineer for the Riter-Conley Manufacturing Company and in this capacity his reputation became world wide. He designed and built furnaces, steel works, oil refineries, gasometers, buildings, bridges

and other steel constructions. He was instrumental in making this concern one of the greatest of its kind in the world. In 1886 he united with S. T. and C. H. Wellman, to form the Wellman-Seaver Engineering Company, later known as the Wellman-Seaver-Morgan Company. While with this company he was vice-president and chairman of the board, and was a director at the time of his death. In 1906 Mr. Seaver associated himself with J. E. Moore, consulting and contracting engineers, and opened offices in Cleveland, O. He continued in this work until his death.

Mr. Seaver's engineering experience was most varied, including the civil, mechanical, marine and mining branches of the profession. While in Buffalo he designed and built many large marine engines, among the most prominent of which was that for the steamer Great Western a remarkable vessel at that time. He designed and built the first gantry crane used in this country and also the first steel cars. He was authority on structural steel construction of all kinds, as well as material handling, steel and iron manufacturing and coke oven machinery, and compiled the first standard steel railroad bridge specification, which was adopted by some of the large railroad companies.

Mr. Seaver was a member of the American Society of Civil Engineers, the Cleveland Engineering Society, and of many business and social organizations.

OLIN AMES STRANAHAN

Olin Ames Stranahan who was born in Litchfield, Ohio, on July 18, 1866, died in New York City on September 8, 1911. Mr. Stranahan received his education at the Case School of Applied Science in Cleveland and at Cornell University, from which he was graduated in 1890. Upon graduation he entered the employ of Westinghouse, Church, Kerr & Company at Chicago, working up through their various departments to the position of chief engineer of their Chicago office. When the British Westinghouse Electric and Manufacturing Company, Ltd., was formed in 1900, Mr. Stranahan was placed in charge of their engine business, resigning in 1905 to accept the position of manager of the power department of the Allis-Chalmers Company, Milwaukee, Wis., shortly afterward becoming general sales manager. In 1907 he joined the Westinghouse Machine Company with headquarters in New York, having charge of gas engine sales and special power installation, and two years later took over the

charge of the export machinery department of the John Deere Export Company of New York. At the time of his death he was general manager of the General Reduction, Gas and By-Products Company of the same city.

Mr. Stranahan specialized during most of his professional life in gas engines and was the owner of several valuable patents applying to gas engines and producers, and had a wide acquaintance in his own country and abroad. He was a member of the Western Society of Engineers, the Engineers Club of New York, and the Engineers Technical and Chicago Athletic Clubs of Chicago.

JOHN TURNER

John Turner was born May 8, 1846 at Cornwall, England, and six years later moved with his family to this country, settling in Michigan where his father became a mine contractor. He received his early education in the schools of Dover, N. J., and in 1863 entered the machine shop of Hewes & Phillips, Newark, N. J., where he served his apprenticeship and later became erecting engineer. In this capacity he was associated with the Watts Campbell Company, Newark, after which he became superintendent of shops for Boone & Perez, Brooklyn, N.Y. From 1874 to 1885 he acted as chief engineer at the State Asylum, Morris Plains, N. J., leaving there to take charge in the same capacity of the Peerless Steam Power Company. In 1887 he became chief engineer of the College of Physicians and Surgeons, New York, and in 1890 superintendent of central station construction for the General Electric Company, having entire charge of the construction of the underground system for the Cincinnati Edison Station and the Cleveland General Electric Company. In 1894 he became supervising engineer on the construction of the American Surety Building and later became its chief engineer for a period of three years. Subsequently he was associated with Sanderson & Porter, contracting and consulting engineers, for whom he built the electric light and railway plants at Far Rockaway, and the lighting plants at Tuxedo and Peekskill, N. Y. In 1901 he became superintendent and chief engineer of the American Bank Note Company, and from 1907 to the time of his death on September 12, 1911, acted in the same capacity at the Bishop Building.

EVERETT HANSON WHITE

Everett Hanson White was born May 26, 1872 at Barre, Mass. He received his early education in the public schools of his native town and in 1888 entered the employ of the Putnam Tool Company, Fitchburg, Mass., taking three years' special training in the shop. He became connected with the Houston Electric Company and later accepted a position with the Wire Goods Company of Worcester, Mass., and with Plummer, Ham and Richardson of the same place. In 1899 he entered the employ of the Locomobile Company of America as draftsman, and at the end of eight months was promoted to the position of chief draftsman at the Bridgeport, Conn. factory where he also had charge of the experimental department. He designed their 1902 Touring Carriages and Worcester engine which developed $7\frac{1}{2}$ h.p. and weighed but 45 lb. In 1903 Mr. White accepted a position with the Eaton, Cole and Burnham Company, of Bridgeport, later controlled by the Crane Valve Company of Chicago, and was then made consulting engineer of the company. In 1905 he entered Sheffield Scientific School of Yale University and completed the four years' course in one year. He died January 12, 1911, at Bridgeport, Conn.

DWIGHT B. WILSON

Col. Dwight B. Wilson, who died at Denver, Colo., on March 7, 1911, was born at Lewiston, Me., on May 5, 1848, and received his technical education at the Holmes Commercial School in Boston, Mass. During the Civil War he served under General B. F. Butler as a private. In 1880 he removed to Colorado where he engaged in construction work, and in 1895 was appointed commissioner of highways in Denver. In 1899 he entered the employ of the Denver City Tramway Company, as superintendent of power. One of his notable achievements during the twelve years of his service was the erection of a new power station costing $1,500,000.

Mr. Wilson was a member of the National Association of Stationary Engineers.

EDWARD B. YARYAN

Edward B. Yaryan who was born in Toledo, Ohio on April 10, 1881, died in that city on April 28, 1911. Mr. Yaryan began his professional career in 1896 with the Toledo Heating and Lighting Company as a steam engineer. From 1900–1902 he was manager

of a central station at Evanston, Ill., and was at the same time employed as superintendent of construction for the Oak Park Yaryan Company of Oak Park, Ill. The following year he re-entered the service of the Toledo Heating and Lighting Company as its superintendent, leaving there in 1906 to become vice-president of the Maclaren & Sprague Lumber Company of Toledo. In 1908–1909 he became superintendent of the Yaryan Process Company of the same city, and in the following year of the Yaryan Naval Stores Company, in the service of which latter firm he later went to Gulfport, Miss., where he remained until shortly before his death.

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